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Research into a method of calculating the stopping distance in the case of goods trains retarded by continuous compressed air brakes, with progressive charging of the brake cylinders,

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PRELIMINARY OBSERVATIONS.

When one examines the mode of operation of the triple valve as fitted to goods trains, it will be noticed that in determining the stopping distances of trains of this kind, it is necessary to consider two successive braking periods, namely :

1) A first period of known length during which the brake cylinders are charged. During this period the total effort (force) of application of the brake shoes increases with time, rapidly at first and increasing progressively, up to a maximum which is reached at the end of the period.

2) A second period, during which the maximum pressure arising in the brake cylinders is maintained and during which braking takes place at constant pressure.

Under these conditions and knowing :

1. the weight of the train and its make-up;

2. the weight and type of locomotive hauling the train;

3. the maximum braking force due to the united braking force of the brake shoes of the waggons braked in the train (including locomotives);

4. the required further effort to be reached;

5. the gradient of that section of the line on which the braking takes place;

6. the speed of the train at the commencement of application of the brakes;

the problem is to ascertain :

1) the speed of the train at the completion of the period of charging the brake cylinders;

2) the distance covered by the train during this initial period;

3) the time elapsing until the train is brought to rest (second period);

4) the distance covered by the train during this period.

* * *

Let :

1) P = a constant = the weight of the whole train (including the locomotive) in tons;

2) P′ = a constant = the weight of the train of waggons, in tons;

3) M = a constant = the mass of the whole train $= \frac{1\,000\ P}{g}$, in which the constant g is the acceleration (9 m. 81 per second²) due to gravity;

4) η = a constant = the coefficient of increase of mass M, taking into account the inertia of gyration of the revolving masses in the train;

5) Q = a constant = the value of the maximum effort due to the application of all the brake shoes on the whole of the vehicles braked in the train, expressed in kgr.;

6) T = a constant = the period of time after which this effort is realised, such period being measured from the start of braking and expressed in seconds;

7) i = a constant = the gradient of the section of the line, expressed in mm. per metre ($+ i$ standing for up gradients and $- i$ for falling gradients).

In addition let :

1) the variable v = speed of the train in m./sec., so that $v = \frac{V}{3.6}$ and therefore $V = 3.6v$, when V stands for the same speed but expressed in km. per hour;

2) the variable t = time in seconds;

3) the variable q = the instantaneous magnitude of the force applied to the brake shoes in any one of the braked vehicles, expressed in kgr. and varying from zero to a maximum during the period of charging, namely :

q_1 for the first braked vehicle;

q_2 for the second braked vehicle;

q_3 for the third braked vehicle; and so on...

. . . .

q_m for the m^{th} braked vehicle;

4) the variable $\Sigma q = q_1 + q_2 + q_3 + \ldots\ldots\ldots\ldots q_m$ = the instantaneous magnitude, during the same period, of the total force applied by the brake shoes on the whole of the braked vehicles, likewise expressed in kgr. and varying from zero to the maximum of Q kgr. as already indicated above;

5) the variable f = the coefficient of friction between the brake blocks and the tyres of the vehicles having brakes in the train;

6) the variable r = specific (*) resistance of the train, expressed in kgr. per ton weight (see Appendix I);

7) the variable R = total (*) resistance of the locomotives hauling the train, expressed in kgr. (see Appendix II).

Finally let :

v_A = speed of train at commencement of braking, speed known and expressed in m./sec.;

v_B = speed of train at end of charging period, speed to be reckoned and expressed in m./sec.;

L = distance travelled by the train during the same period, distance to be reckoned and expressed in m.;

(*) i.e. resistance to running plus resistance of the air.

T′ = time elapsed during the second period, from commencement of speed v_B up to stoppage of the train, time to be reckoned and expressed in seconds;

L′ = distance covered by the train during the same period, distance to be reckoned and expressed in metres;

so that :

T + T′ = total braking period;

L + L′ = distance covered by the train till it comes to rest, that being the figure which it is sought to ascertain.

Due to the kindness of the Swedish Brake Cy. S.A.B., we have received diagrams A and B, figure 1, drawn by that firm at their testing laboratory at Malmoë.

These two diagrams (Fig. 1) show, by way of example, for the Westinghouse brake as supplied for freight trains, the speed of operation during the period of charging the brake cylinders, from the initial application of the brakes until the maximum pressure is attained, the curves expressing graphically the following functions :

1) $q_1 = \Psi_1\ (t)$ for one waggon (A) at the head of a string of 50 waggons with brake cylinders set for a stroke of 100 mm. (diagram A);

2) $q_2 = \Psi_2\ (t)$ for a waggon (B) at the tail of a string af 50 waggons, with stroke of piston set to 200 mm. (diagram B).

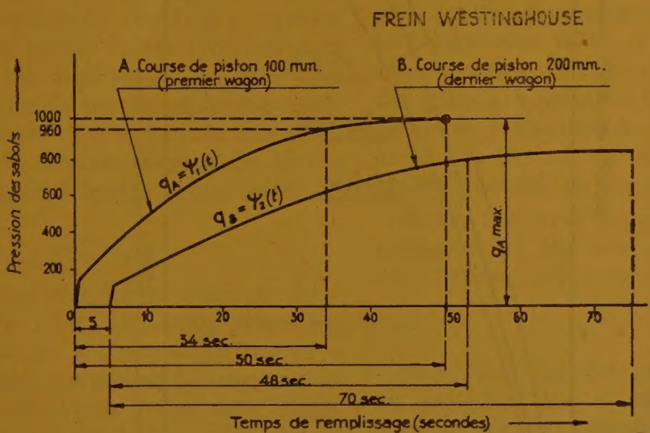

Fig. 1.

Explanation of French terms :

Frein Westinghouse = Westinghouse brake. — A. Course du piston 100 mm. (premier wagon) = A. Piston-stroke 100 mm. (first waggon). — B. Course de piston 200 mm. (dernier wagon) = B. Piston-stroke 200 mm. (last waggon). — Pression des sabots = Pressure of the brake shoes. — Temps de remplissage (secondes) — Period necessary for charging the brake cylinders (seconds).

* * *

Preliminary examination of an analytical form of the equation $\Sigma q = \Psi(t)$ which will indicate the instantaneous magnitude of the total braking force applied, namely (Σq) as a function of the time (t) covering the period necessary for charging the brake cylinders.

The conclusions which may be drawn from an examination of these two diagrams may be stated as follows :

1) The speed of operation shown in diagram waggon A is quite different from that indicated in the diagram referring to waggon B. This difference is not due to the fact that waggon B is at the tail end of the train, but exclusively to the perceptible difference in the setting of the strokes of the brake cylinders. Actually, owing to the action of the auxiliary reservoir forming part of the triple valves on goods trains, the position of the waggon in the train has scarcely any influence on the rate of application shown by the diagram under consideration.

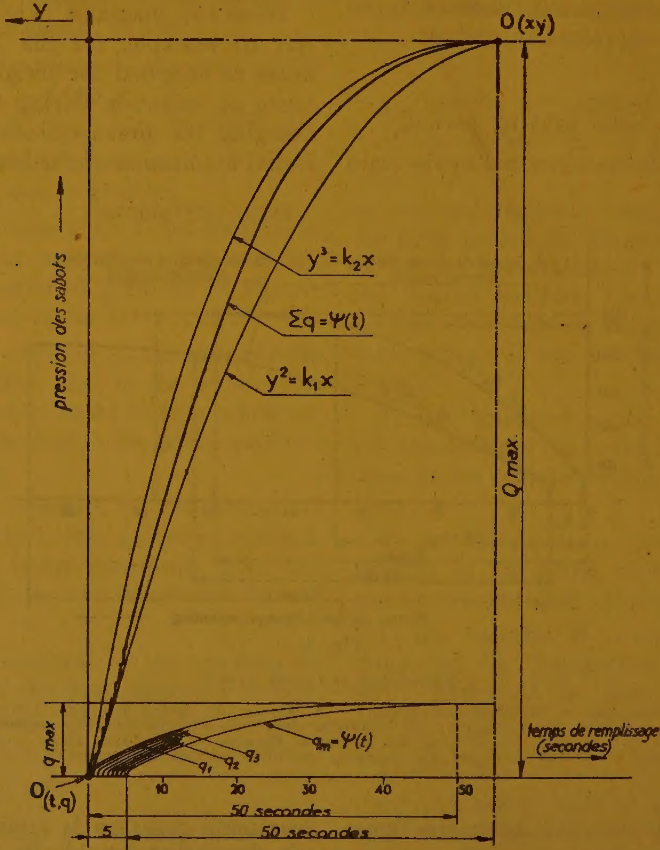

Fig. 2.

Explanation of French terms:

Pression des sabots = Pressure of the brake shoes. — Temps de remplissage (secondes) = Period necessary for charging the brake cylinders (seconds).

2) One can thus, by interpolation, trace the rate of operation proper to this diagram for any waggon brake on the train,

the movement of the brake cylinder pistons being taken as adjusted to a given value (between 100 and 200 mm.).

3) The period of propagation required by the wave of diminishing pressure in the train pipe was 5 seconds for a train of 50 waggons, say $\frac{1}{10}$th. of a second per waggon.

4) The period required for charging the brake cylinders of the waggon A with piston stroke set to 100 mm. was practically 34 seconds for realising a pressure of the brake blocks on the tyres equal to 95 % of the maximum pressure, and about 50 seconds for reaching the maximum pressure. These periods amounted to 48 and 70 seconds respectively in the case of waggon B having piston travel set to 200 mm.

* * *

Using these data, we were able to draw the diagram (Fig. 2) having reference, for example, to a train consisting of a braked locomotive and of 54 waggons, of which 9 with brakes fitted were evenly distributed over the length of the train, at the rate of 1 braked waggon to every 6 waggons, 5 of which will be without brakes.

As already stated the case for the braked locomotive and for each of the 9 waggons, will be in accordance with the « pressure-time » diagram A shown in Fig. 1 (stroke of the pistons set to 100 mm.).

Under these conditions :

1) the various « pressure-time » diagrams will be spaced out in time, over $6 \times \frac{1}{10}$ second $= \frac{6}{10}$ second or say approximately one ½ second. This spacing is shown horizontally below the diagram, fig. 2.

2) The progressive increase of total pressure of the brake blocks on the tyres of the whole of the braked vehicles comprising the train, extends from the commencement of braking until the maximum pressure is reached, over a period of time (T) equal to $50'' + \left(54 \times \frac{1}{10}\right) = 50 + 5.4 = 55.4$ seconds, say 55 seconds in round figures.

And this total pressure of the brake blocks on the tyres, which rises from zero (at the commencement of braking) to a maximum value of Q kgr. (reached after a period of T seconds) provides, at any instant, a pressure of Σq kgr. = Σ times the instantaneous pressures of the brake blocks on the total number of braked vehicles in the train.

Hence, the curve $\Sigma q = \Psi(t)$ in fig. 2 represents, in the case under consideration, the law which we wish to define.

* * *

There remains the need to attribute an analytical law to this curve. It is possible, without introducing an appreciable error, to assimilate the curve $\Sigma q = \Psi(t)$, obtained as shown above, to a function $y^n = Kx$, the original of the co-ordinates (x, y) being taken at the point $O_{x,y}$ of the curve $\Sigma q = \Psi(t)$ in which $\Sigma q = Q$ (Fig. 2).

To do this we must change over from the system of co-ordinates x, y to that of t, Σq.

A simple operation of transforming co-ordinates allows us to write :

$$\begin{cases} t = T - y, \text{ whence } y = T - t \\ \Sigma q = Q - x, \text{ whence } x = Q - \Sigma q \end{cases}$$

so that the equation $y^n = Kx$ will take the form,

$$(T - t)^n = K(Q - q)$$

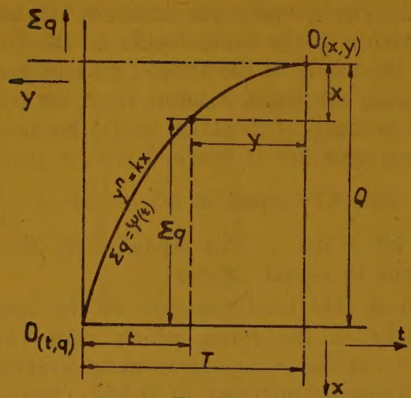

Fig. 3.

For $t = 0$, we have : $q = 0$, from which $T^n = KQ$ and therefore : $K = \frac{T^n}{Q}$, so that the equation under consideration becomes :

$$(T - t)^n = \frac{T^n(Q - \Sigma q)}{Q}$$

or in other words :

$$\Sigma q = Q - \frac{Q(T - t)^n}{T^n}$$

We then verify that :

For $t = 0$, $\Sigma q = Q - \frac{QT^n}{T^n} =$

$Q - Q = 0$

For : $t = T, \Sigma q = Q - \frac{Q(T - T)^n}{T^n}$

$$= Q - \frac{0}{T^n}$$

$$= Q - 0$$

$$= Q$$

As regards the value to be given to the exponent n, we can ascertain from an examination of the form of the curve $\Sigma q = \Psi(t)$ of Fig. 2 that this curve is practically half way between a parabola of the second degree and one of the third degree. The function $y^n = Kx$ will therefore lie between $y^2 = K_1x$ and $y^3 = K_2x$, so that the value of n will lie between 2 and 3. In the case of the curve under consideration (Fig. 2) it appears that the value of n may be taken to be equal in this case to 2.5, for example.

In a particular case it will suffice to trace between the points $0_{x,y}$ and $0_{t,\Sigma q}$ a group of parabolas $y^n = Kx$, having the values of n lying between 2 and 3, and to ascertain, by comparison between each of the parabolas and the real curve $\Sigma q = \Psi(t)$, the value of n which comes nearest to the true value.

FIRST PART OF THE OPERATION.

Calculation of the stopping distance in the simple case where the train, during the whole period of braking, moves in a section of the line having a continuous gradient (i).

I. FIRST PERIOD.

Braking operation during the period necessary for charging the brake cylinders.

A. *Examination of the general relationship between speed, distance travelled and time.*

Considered under the most general conditions, the problem of examining the relations existing between these three factors, during the above-mentioned period, presents itself as indicated below :

We set :

1. Two time limits $t = \Theta_1$, and $t = \Theta_2$ included between extreme limits $t = 0$ and $t = T$;

2. The speed v_1 corresponding to the lower time limit $t = \Theta_1$;

3. The gradient i of that section of the line under consideration.

The object is to find :

1. The speed v_2 corresponding to the upper time limit $t = \Theta_2$;

2. The distance (l) travelled during the time interval $\Theta_2 - \Theta_1$.

*
* *

Taking into account the fact that the coefficient f is affected not only by the speed, but also by *duration* of braking and by the intensity of the *specific pressure* of the blocks on the wheel tyres, and observing that the pressure varies from one waggon to another and from one instant to another (since during the period in question, the pressure q increases with the time) the differential equation appertaining to the movement, and valid between the time limits Θ_1 and Θ_2 should read :

$$M\eta \frac{dv}{dt} = - \Big\{ f_1q_1 + f_2q_2 + f_3q_3 + \ldots f_mq_m + rP' + R \pm iP \Big\} . \quad . \ (1)$$

a relationship in which :

f_1 = a function of v, t, q_1, with q_1 being a function of t

f_2 = a function of v, t, q_2, with q_2 being a function of t

. . . .

f_m = a function of v, t, q_m, with q_m being a function of t

so that the above equation becomes so complex that a solution cannot be undertaken.

In order to get over this difficulty, we propose the following simplifications in the calculations :

1. That the parameters for the rule $f = \varphi(v)$ which is commonly used, should correspond to a certain degree to mean values, which implicitly take into account the effect of the duration and of the diversity of the specific pressures, so that we can write :

$$f\Sigma q = f_1q_1 + f_2q_2 + \ldots \ldots f_mq_m.$$

Under these conditions the differential equation (1) can be set out more simply:

$$M\eta \frac{dv}{dt} = - \Big\{ f\,\Sigma q + rP' + R \pm iP \Big\} . \ . \ (2)$$

a relationship in which :

$f = F(v)$, $\Sigma q = \Psi(t)$, $r = \varphi\ (v)$, $R = \Phi\ (v)$

2. That the variables, f, r and R can likewise be expressed as functions of time, so that the foregoing equation (2) can finally take the form :

$$M\eta \frac{dv}{dt} = - \Big\{ \Phi\ (t) \Big\}$$

Under these circumstances let us examine the following values of the variables f, r and R :

f_1 = special value when $v = v_1$

f_2 = special value when $v = v_2$

r_1 = special value when $v = v_1$

r_2 = special value when $v = v_2$

R_1 = special value when $v = v_1$

R_2 = special value when $v = v_2$

By accepting the approximation of a linear law as a function of time between the limits of speeds v_1 and v_2 which correspond respectively to the limits Θ_1 and Θ_2 we can write :

$$f = f_1 + (t - \Theta_1)\alpha$$
$$= f_1 + \alpha t - \alpha\Theta_1$$
$$r = r_1 - (t - \Theta_1)\beta$$
$$= r_1 - \beta t + \beta\Theta_1$$
$$R = R_1 - (t - \Theta_1)\gamma$$
$$= R_1 - \gamma t + \gamma\Theta_1$$

expressions in which :

$$\alpha = \frac{f_2 - f_1}{\Theta_2 - \Theta_1}$$

$$\beta = \frac{r_1 - r_2}{\Theta_2 - \Theta_1}$$

$$\gamma = \frac{R_1 - R_2}{\Theta_2 - \Theta_1}$$

In these conditions, the differential equation (2), valid between the limits $t = \Theta_1$, and $t = \Theta_2$ may be written :

$$M\eta \frac{dv}{dt} = - \{ (f_1 + \alpha t - \alpha\Theta_1) \Sigma q + (r_1 - \beta t + \beta\Theta_1)P' + R_1 - \gamma t + \gamma\Theta_1 \pm iP \}$$

or by substituting Σq by the value : $Q - \dfrac{Q(T - t)^n}{T^n}$ which was deduced above :

$$M\eta \frac{dv}{dt} = - \left\{ \left[f_1 + \alpha t - \alpha\Theta_1 \right] \left[Q - \frac{Q(T - t)^n}{T^n} \right] + (r_1 - \beta t + \beta\Theta_1)P' + (R_1 - \gamma t + \gamma\Theta_1) \pm iP \right\} \quad . . . (3)$$

* * *

In this equation however we must point out that the angular coefficients α, β and γ are unknown since they depend on f_1, r_1, R_1, and on f_2, r_2, R_2, these three last values are themselves unknown as they are determined by the speed v_2 which is just what we are trying to find.

Consequently we have to have recourse to a method of calculation by successive approximations, as for example :

1) To seek a first approximate value of the speed v_2 treating, as a first approximation, the factors f, r, and R as constants and proposing a value, respectively equal to f_1, r_1, R_1 which will amount to momentarily making $\alpha = 0$, $\beta = 0$ and $\gamma = 0$ in equation (3). The value v_2' (approx. value) provided by this last allows us to deduce the following :

$$\begin{cases} f_2' = a - bv_2' \\ r_2' = \text{value of } r \text{ for } v = v_2' \quad \text{(Appendix I)} \\ R_2' = \text{value of R for } v = v_2' \quad \text{(Appendix II)} \end{cases}$$

from which it follows that :

$$\begin{cases} \alpha' = \dfrac{f_2' - f_1}{\Theta_2 - \Theta_1} \\ \beta' = \dfrac{r_1 - r_2'}{\Theta_2 - \Theta_1} \\ \gamma' = \dfrac{R_1 - R_2'}{\Theta_2 - \Theta_1} \end{cases}$$

2) To introduce the values α', β', γ', found in this way, in the equation (3), which consequently provides us with a value more closely approximated to v_2 from which it is possible to deduce values α'', β'', γ'', which are still more closely approximated.

3) To introduce in turn these values α'', β'', γ'', in the equation (3), thus providing a value still more closely approximated to v_2, and so on.

In what follows we shall restrict ourselves to a second approximation.

* * *

Let us now proceed with equation (3). This may in turn be written :

$$M\eta \frac{dv}{dt} = -\Big\{ f_1Q - \frac{f_1Q(T-t)^n}{T^n} + \alpha Qt - \frac{\alpha Q(T-t)^n t}{T^n} - \alpha\,\Theta_1 Q$$

$$+ \frac{\alpha\,\Theta_1 Q(T-t)^n}{T^n} + r_1P' - \beta P't + \beta\,\Theta_1 P' + R_1 - \gamma t + \gamma\,\Theta_1 \pm iP \Big\}$$

$$= -\Big\{ \frac{\alpha\,\Theta_1 Q(T-t)^n}{T^n} - \frac{f_1Q(T-t)^n}{T^n} - \frac{\alpha Q(T-t)^n t}{T^n} + \alpha Qt - \beta P't$$

$$- \gamma t + f_1Q + r_1P' + R_1 - \alpha\,\Theta_1 Q + \beta\,\Theta_1 P' + \gamma\,\Theta_1 \pm iP \Big\}$$

$$= -\Big\{ \left[\frac{\alpha\,\Theta_1 Q}{T^n} - \frac{f_1Q}{T^n}\right](T-t)^n - \frac{\alpha Q}{T^n}(T-t)^n t + (\alpha Q - \beta P' - \gamma)t$$

$$+ f_1Q + r_1P' + R_1 - \alpha\Theta_1 Q + \beta\Theta_1 P' + \gamma\Theta_1 \pm iP \Big\}$$

so that by writing :

$$A = \frac{\alpha\,\Theta_1 Q}{T^n} - \frac{f_1Q}{T^n}$$

$$B = \frac{\alpha Q}{T^n}$$

$$C = \alpha Q - \beta P' - \gamma$$

$$D = f_1Q + r_1P' + R_1 - \alpha\Theta_1 Q + \beta\Theta_1 P' + \gamma\Theta_1 \pm iP$$

we get :

$$M\eta \frac{dv}{dt} = -\Big\{A(T-t)^n - B(T-t)^n t + Ct + D\Big\}$$

whence

$$\frac{dv}{dt} = -\frac{1}{M\eta}\Big\{A(T-t)^n - B(T-t)^n t + Ct + D\Big\}$$

$$= -\frac{g}{1000P\eta}\Big\{A(T-t)^n - B(T-t)^n t + Ct + D\Big\}$$

or by substituting $E = \dfrac{g}{1000P\eta}$:

$$\frac{dv}{dt} = -E\Big\{A(T-t)^n - B(T-t)^n t + Ct + D\Big\}$$

whence

$$dv = -E\Big\{A(T-t)^n dt - B(T-t)^n t\,dt + Ct\,dt + D\,dt\Big\}$$

and

$$\int_{v_1}^{v_2} dv = -E \left\{ A \int_{\Theta_1}^{\Theta_2} (T - t)^n dt - B \int_{\Theta_1}^{\Theta_2} (T - t)^n t dt + C \int_{\Theta_1}^{\Theta_2} t dt + D \int_{\Theta_1}^{\Theta_2} dt \right\}$$

$$= -E \left\{ -\frac{A}{n+1} \Big[(T - t)^{n+1} \Big]_{\Theta_1}^{\Theta_2} + \frac{BT}{n+1} \Big[(T - t)^{n+1} \Big]_{\Theta_1}^{\Theta_2} \right.$$

$$\left. - \frac{B}{n+2} \Big[(T - t)^{n+2} \Big]_{\Theta_1}^{\Theta_2} + \frac{C}{2} \Big[t^2 \Big]_{\Theta_1}^{\Theta_2} + D \Big[t \Big]_{\Theta_1}^{\Theta_2} \right\}$$

$$= -E \left\{ \frac{BT - A}{n+1} \Big[(T - t)^{n+1} \Big]_{\Theta_1}^{\Theta_2} - \frac{B}{n+2} \Big[(T - t)^{n+2} \Big]_{\Theta_1}^{\Theta_2} \right.$$

$$\left. + \frac{C}{2} \Big[t^2 \Big]_{\Theta_1}^{\Theta_2} + D \Big[t \Big]_{\Theta_1}^{\Theta_2} \right\}$$

or by substituting :

$$F = \frac{BT - A}{n+1}$$

$$G = \frac{B}{n+2}$$

$$H = \frac{C}{2}$$

$$\int_{v_1}^{v^2} dv = -E \left\{ F \Big[(T - t)^{n+1} \Big]_{\Theta_1}^{\Theta_2} - G \Big[(T - t)^{n+2} \Big]_{\Theta_1}^{\Theta_2} + H \Big[t^2 \Big]_{\Theta_1}^{\Theta_2} + D \Big[t \Big]_{\Theta_1}^{\Theta_2} \right\}$$

or :

$$v_2 - v_1 = -E \left\{ F \Big[(T - t)^{n+1} \Big]_{\Theta_1}^{\Theta_2} - G \Big[(T - t)^{n+2} \Big]_{\Theta_1}^{\Theta_2} + H \Big[t^2 \Big]_{\Theta_1}^{\Theta_2} + D \Big[t \Big]_{\Theta_1}^{\Theta_2} \right\}$$

whence

$$v_2 = v_1 - E \left\{ F \Big[(T - t)^{n+1} \Big]_{\Theta_1}^{\Theta_2} - G \Big[(T - t)^{n+2} \Big]_{\Theta_1}^{\Theta_2} + H \Big[t^2 \Big]_{\Theta_1}^{\Theta_2} + D \Big[t \Big]_{\Theta_1}^{\Theta_2} \right\}$$

or, finally, after completing the calculations :

$$v = v_1 - \mathrm{E} \Big\{ \mathrm{G}(\mathrm{T} - \Theta_1)^{n+2} - \mathrm{G}(\mathrm{T} - \Theta_2)^{n+2} + \mathrm{F}(\mathrm{T} - \Theta_2)^{n+1} - \mathrm{F}(\mathrm{T} - \Theta_1)^{n+1} + \mathrm{H}\,\Theta_2^2 - \mathrm{H}\,\Theta_1^2 + \mathrm{D}\,\Theta_2 - \mathrm{D}\,\Theta_1 \Big\} \quad . \; . \; . \quad (4)$$

Assuming that $v_2 = v$ and $\Theta_2 = t$, the above relationship (4) becomes :

$$v = v_1 - \mathrm{E} \Big\{ \mathrm{G}(\mathrm{T} - \Theta_1)^{n+2} - \mathrm{G}(\mathrm{T} - t)^{n+2} + \mathrm{F}(\mathrm{T} - t)^{n+1} - \mathrm{F}(\mathrm{T} - \Theta_1)^{n+1} + \mathrm{H}t^2 - \mathrm{H}\,\Theta_1^2 + \mathrm{D}t - \mathrm{D}\,\Theta_1 \Big\}$$

whence :

$$vdt = v_1dt - \mathrm{E} \Big\{ \mathrm{G}(\mathrm{T} - \Theta_1)^{n+2}dt - \mathrm{G}(\mathrm{T} - t)^{n+2}dt + \mathrm{F}(\mathrm{T} - t)^{n+1}dt - \mathrm{F}(\mathrm{T} - \Theta_1)^{n+1}dt + \mathrm{H}t^2dt - \mathrm{H}\,\Theta_1^2dt + \mathrm{D}tdt - \mathrm{D}\,\Theta_1dt \Big\}$$

so that :

$$l = \int_{\Theta_1}^{\Theta_2} vdt$$

$$= v_1 \int_{\Theta_1}^{\Theta_2} dt - \mathrm{E} \Big\{ \mathrm{G}(\mathrm{T} - \Theta_1)^{n+2} \int_{\Theta_1}^{\Theta_2} dt - \mathrm{G} \int_{\Theta_1}^{\Theta_2} (\mathrm{T} - t)^{n+2}dt + \mathrm{F} \int_{\Theta_1}^{\Theta_2} (\mathrm{T} - t)^{n+1}dt - \mathrm{F}(\mathrm{T} - \Theta_1)^{n+1} \int_{\Theta_1}^{\Theta_2} dt + \mathrm{H} \int_{\Theta_1}^{\Theta_2} t^2dt - \mathrm{H}\,\Theta_1^2 \int_{\Theta_1}^{\Theta_2} dt + \mathrm{D} \int_{\Theta_1}^{\Theta_2} tdt - \mathrm{D}\Theta_1 \int_{\Theta_1}^{\Theta_2} dt \Big\}$$

or, after completing calculations :

$$l = v_1(\Theta_2 - \Theta_1) - \mathrm{E} \Big\{ \frac{\mathrm{G}}{n+3}\left[(\mathrm{T} - \Theta_2)^{n+3} - (\mathrm{T} - \Theta_1)^{n+3}\right] - \frac{\mathrm{F}}{n+2}\left[(\mathrm{T} - \Theta_2)^{n+2} - (\mathrm{T} - \Theta_1)^{n+2}\right] + \frac{\mathrm{H}}{3}\left[\Theta_2^3 - \Theta_1^3\right] + \frac{\mathrm{D}}{2}\left[\Theta_2^2 - \Theta_1^2\right] + \left[\mathrm{G}(\mathrm{T} - \Theta_1)^{n+2} - \mathrm{F}(\mathrm{T} - \Theta_1)^{n+1} - \mathrm{H}\,\Theta_1^2 - \mathrm{D}\Theta_1\right]\left[\Theta_2 - \Theta_1\right] \Big\}$$

By substituting : $J = G(T - \Theta_1)^{n+2} - F(T - \Theta_1)^{n+1} - H\Theta_1^2 - D\Theta_1$, we finally obtain :

$$l = v_1 (\Theta_2 - \Theta_1) - E \left\{ \frac{G(T - \Theta_2)^{n+3}}{n+3} - \frac{G(T - \Theta_1)^{n+3}}{n+3} \right.$$

$$+ \frac{F(T - \Theta_1)^{n+2}}{n+2} - \frac{F(T - \Theta_2)^{n+2}}{n+2} + \frac{H}{3}\Theta_2^3 - \frac{H}{3}\Theta_1^3 + \frac{D}{2}\Theta_2^2$$

$$\left. - \frac{D}{2}\Theta_1^2 + J\Theta_2 - J\Theta_1 \right\} \quad . \quad . \quad . \quad (5)$$

It will be well to note in connection with the two general equations (4) and (5) that :

$$A = \frac{\alpha\Theta_1 Q}{T^n} - \frac{f_1 Q}{T^n}$$

$$B = \frac{\alpha Q}{T^n}$$

$$C = \alpha Q - \beta P' - \gamma$$

$$D = f_1 Q + r_1 P' + R_1 - \alpha\Theta_1 Q + \beta\Theta_1 P' + \gamma\Theta_1 \pm iP$$

$$E = \frac{g}{1000\ P\eta}$$

$$F = \frac{BT - A}{n+1} = \frac{\dfrac{\alpha QT}{T^n} - \dfrac{\alpha\Theta_1 Q}{T^n} + \dfrac{f_1 Q}{T^n}}{n+1}$$

$$= \frac{\alpha Q}{(n+1)T^{n-1}} - \frac{\alpha\Theta_1 Q}{(n+1)T^n} + \frac{f_1 Q}{(n+1)\ T^n}$$

$$G = \frac{B}{n+2} = \frac{\alpha Q}{(n+2)T^n}$$

$$H = \frac{C}{2} = \frac{\alpha Q - \beta P' - \gamma}{2}$$

$$J = G(T - \Theta_1)^{n+2} - F(T - \Theta_1)^{n+1} - H\ \Theta_1^2 - D\ \Theta_1$$

B. *Application of the general laws (4) and (5) bearing on the determination of the speed (v_B) characterizing the end of the charging period and on the determination of the distance (L) travelled during this period.*

FIRST ASSUMPTION.

We assume that during the whole period (T seconds) of charging the brake cylinders, the coefficient of friction f between the brake blocks and the tyres varies as a function of the speed, in accordance with a single linear law f = a — bv *from speed* $v_1 = v_A$ *(commencement of braking) down to speed* $v_2 = v_B$ *(end of charging period).*

On this assumption :

$f_1 = f_A = a - bv_A$ at commencement of braking.
$r_1 = r_A = \varphi(v_A)$ at commencement of braking.
$R_1 = R_A = \Phi(v_A)$ at commencement of braking.

$f_2 = f_B = a - bv_B$ at end of charging period.
$r_2 = r_B = \varphi(v_B)$ at end of charging period.
$R_2 = R_B = \Phi(v_B)$ at end of charging period.

1) *Calculation as a first approximation of the speed* $v_2 = v_B$.

Let us designate by v'_B the value of v_B obtained as a first approximation, by putting $\alpha = 0$, $\beta = 0$, $\gamma = 0$, i.e. by considering for the moment and as a first approximation, the factors f, r and R as constants and applying to them values equal to those of f_A, r_A, R_A which correspond to the commencement of braking.

If then, under these conditions, we introduce into the relationship (4) the values :

$\Theta_1 = 0$	$f_1 = f_A = a - bv_A$	$\alpha = 0$
$\Theta_2 = T$	$r_1 = r_A = \varphi(v_A)$	$\beta = 0$
$v_1 = v_A$	$R_1 = R_A = \Phi(v_B)$	$\gamma = 0$
$v_2 = v'_B$		

so that :

$$A = -\frac{f_A Q}{T^n}$$
$$B = 0$$
$$C = 0$$
$$D = f_A Q + r_A P' + R_A \pm iP$$
$$E = \frac{g}{1000\ P\eta}$$
$$G = 0$$
$$H = 0$$

the said relationship (4) giving us after carrying out the necessary calculations :

$$v'_B = v_A - \frac{gT}{1000\ P\eta}\left\{\frac{nf_A Q}{n+1} + r_A P' + R_A \pm iP\right\}$$

from this we are able to deduce the approximate values of : $f_2 = f_B$, $r_2 = r_B$ and $R_2 = R_B$, namely :

$$f_2 = f_B = a - bv'_B \qquad \alpha = \frac{f_B - f_A}{T}$$

$$r_2 = r = \varphi(v'_B) \qquad \text{whence} \qquad \beta = \frac{r_A - r_B}{T}$$

$$R_2 = R_B = \Phi(v'_B) \qquad \gamma = \frac{R_A - R_B}{T}$$

2) *Calculation as a second approximation of the speed* v_B.

In introducing the following values in the relationship (4) :

$$
\begin{array}{l|l|l}
\Theta_1 = 0 & f_1 = f_A = a - bv_A & \alpha = \dfrac{f_B - f_A}{T} \\
\Theta_2 = T & f_2 = f_B = a - bv'_B & \\
v_1 = v_A & r_1 = r_A = \varphi\,(v_A) & \beta = \dfrac{r_A - r_B}{T} \\
v_2 = v_B & r_2 = r_B = \varphi\,(v'_B) & \\
 & R_1 = R_A = \Phi\,(v_A) & \gamma = \dfrac{R_A - R_B}{T} \\
 & R_2 = R_B = \Phi\,(v'_B) &
\end{array}
$$

with the result that :

$$A = -\frac{f_A Q}{T^n}$$

$$B = \frac{\alpha Q}{T^n}$$

$$C = \alpha Q - \beta P' - \gamma$$

$$D = f_A Q + r_A P' + R_A \pm iP$$

$$E = \frac{g}{1000\,P\eta}$$

$$F = \frac{BT - A}{n+1} = \frac{\alpha Q}{(n+1)T^{n-1}} + \frac{f_A Q}{(n+1)T^n}$$

$$G = \frac{B}{n+2} = \frac{\alpha Q}{(n+1)T^n}$$

$$H = \frac{C}{2} = \frac{\alpha Q - \beta P' - \gamma}{2}$$

the said relationship (4) provides us, after carrying out the necessary calculations, with the equation :

$$v_B = v_A - \frac{gT}{1000\,P\eta}\left\{\frac{nf_A Q}{2(n+2)} + \frac{n(n+3)f_B Q}{2(n+1)(n+2)} + \tfrac{1}{2}(r_A + r_B)P' + \tfrac{1}{2}(R_A + R_B) \pm iP\right\}$$

3) *Calculation of the distance* L *travelled during the period of charging the brake cylinders.*

The relationship (equation 5), by introducing the same values for Θ_1, Θ_2, v_1, α, β, γ, A, B, C, D, E, F, G, H, as also the values $l = L$ and $J = GT^{n+2} - FT^{n+1}$, and after the necessary calculations, gives us :

$$L = v_A T - \frac{gT^2}{1000\,P\eta}\left\{\frac{nf_A Q}{3(n+3)} + \frac{n(n+5)f_B Q}{6(n+2)(n+3)} + \tfrac{1}{3}r_A P' + \tfrac{1}{6}r_B P' + \tfrac{1}{3}R_A + \tfrac{1}{6}R_B \pm \tfrac{1}{2}iP\right\}$$

SECOND ASSUMPTION.

Let us now suppose that in the course of the charging period the coefficient of friction f *of the brake-shoes against the tyres varies, as a function of the speed and in accordance with a hyperbolic law capable of being split up into two linear laws, such that the speed of transition* (v_0) *lies between the speeds* v_A *and* v_B.

In each case, we shall have to consider two successive phases, i.e.

— *A first phase*, governed by the linear law $f = a_1 - b_1 v$ from the speed v_A, which characterizes the commencement of braking, up to the *transition speed* v_0 which is determined by the relationship : $a_1 - b_1 v_0 = a_2 - b_2 v_0$.

— *A second phase* is governed by the second linear law $f = a_2 - b_2 v_0$, from the transition speed v_0 up to the speed v_B which characterizes the termination of the charging period.

In the calculations which follow, let :

f_0 = value of the coefficient f corresponding to the speed of transition (v_0);

r_0 = the specific resistance of the train of vehicles corresponding to this speed;

R_0 = the total resistance of the locomotives corresponding to this same speed.

First phase.

$f_1 = f_A = a_1 - b_1 v_A$ at the commencement of braking.

$r_1 = r_A = \varphi(v_A)$ at the commencement of braking.

$R_1 = R_A = \Phi(v_A)$ at the commencement of braking.

$f_2 = f_0 = a_1 - b_1 v_0 = a_2 - b_2 v_0$ at the speed of transition v_0.

$r_2 = r_0 = \varphi(v_0)$ at the speed of transition v_0.

$R_2 = R_0 = \Phi(v_0)$ at the speed of transition v_0.

1) *Calculation, as a first approximation of the time* t_1, *elapsed between the commencement of braking and the instant when the specified speed of transition* (v_0) *has been reached.*

Let t_1' be the value of t_1, obtained as a first approximation by putting $\alpha = 0$, $\beta = 0$, $\gamma = 0$. Thereafter, if we introduce into equation (4) the values :

$\Theta_1 = 0$	$f_1 = f_A = a_1 - b_1 v_A$	$\alpha = 0$
$\Theta_2 = t_1'$	$r_1 = r_A = \varphi(v_A)$	$\beta = 0$
$v_1 = v_A$	$R_1 = R_A = \Phi(v_A)$	$\gamma = 0$
$v_2 = v_0$		

so that

$$A = -\frac{f_A Q}{T^n}$$

$$B = 0$$

$$C = 0$$

$$D = f_A Q + r_A P' + R_A \pm iP$$

$$E = \frac{g}{1000\ P\eta}$$

$$F = \frac{BT - A}{n+1} = \frac{f_A Q}{(n+1)T^n}$$

$$G = 0$$

$$H = 0$$

the said equation (4) reads :

$$\begin{aligned} v_0 &= v_A - E\left\{F(T - t_1')^{n+1} - FT^{n+1} + Dt_1'\right\} \\ &= v_A - EF(T - t_1')^{n+1} + EFT^{n+1} - EDt_1' \end{aligned}$$

By putting $T - t_1' = X$ so that $t_1' = T - X$, we get :

$$v_0 = v_A - EFX^{n+1} + EFT^{n+1} - ED(T - X)$$

or :

$$v_0 = v_A - EFX^{n+1} + EFT^{n+1} - EDT + EDX$$

or :

$$EFX^{n+1} - EDX - EFT^{n+1} + EDT - v_A + v_0 = 0$$

or :

$$EFX^{n+1} - EDX - EFT^{n+1} + EDT - (v_A - v_0) = 0$$

an equation of the form of

$$a.X^{n+1} + b.X + c = 0$$

the graphic solution of which gives us the value of X and consequently, $t_1' = T - X$, the parameters a, b and c working out to :

$$\begin{aligned} a &= EF \\ b &= -ED \\ c &= -EFT^{n+1} + EDT - (v_A - v_0) \end{aligned}$$

using the values of A, D, E, F, indicated above.

Having thus determined t_1' in a first approximation, we find that :

$$\alpha_1 = \frac{f_0 - f_A}{t_1'}$$

$$\beta_1 = \frac{r_A - r_0}{t_1'}$$

$$\gamma_1 = \frac{R_A - R_0}{t_1'}$$

2) *Determination of* t_1 *in a second approximation.*

If in equation (4) we now introduce the values $\Theta_1 = 0$, $\Theta_2 = t_1$, $v_1 = v_A$, $v_2 = v_0$ and likewise that $\alpha = \alpha_1$, $\beta = \beta_1$ and $\gamma = \gamma_1$, this equation gives us a second approximation :

$$v_0 = v_A - E\left\{GT^{n+2} - G(T - t_1)^{n+2} + F(T - t_1)^{n+1} - FT^{n+1} + H_1t^2 + Dt_1\right\}$$

Putting $T - t_1 = t_2$ so that $t_1 = T - t_2$; we get after necessary calculations:

$$EGt_2^{n+2} - EFt_2^{n+1} - Ht_2^2 + (2HT + D)t_2 + EGT^{n+2} + EFT^{n+1} - HT^2 - DT - (v_A - v_0) = 0$$

which equation is in the form of

$$a.t_2^{n+2} + b.t_2^{n+1} + c.t_2^2 + d.t_2 + e = 0$$

which when solved graphically, gives us the value of t_2 and in consequence $t_1 = T - t_2$, the parameters a, b, c, d, e, having the values :

$$a = EG$$
$$b = - EF$$
$$c = - H$$
$$d = 2HT + D$$
$$e = EGT^{n+2} + EFT^{n+1} - HT^2 - DT - (v_A - v_0)$$

with the following values :

$$A = - \frac{f_A Q}{T^n}$$

$$B = \frac{\alpha_1 Q}{T^n}$$

$$C = \alpha_1 Q - \beta_1 P' - \gamma_1$$

$$D = f_A Q + r_A P' + R_A \pm iP$$

$$E = \frac{g}{1000 \; P r_i}$$

$$F = \frac{BT - A}{n + 1} = \frac{\alpha_1 Q}{(n+1)T^{n-1}} + \frac{f_A Q}{(n+1)T^n}$$

$$G = \frac{B}{n+2} = \frac{\alpha_1 Q}{(n+2)T^n}$$

$$H = \frac{C}{2} = \frac{\alpha_1 Q - \beta_1 P' - \gamma_1}{2}$$

3) *Calculation of the distance* l_1 *travelled in the time* t_1.

If finally, we introduce into the equation (5), the values $\Theta_1 = 0$, $\Theta_2 = t_1$, $v_1 = v_A$, as well as the value $l = l_1$, this equation gives us :

$$l_1 = v_A t_1 - E \left\{ \frac{G(T - t_1)^{n+3}}{n+3} - \frac{GT^{n+3}}{n+3} + \frac{FT^{n+2}}{n+2} - \frac{F(T - t_1)^{n+2}}{n+2} + \frac{H}{3} t_1^3 + \frac{D}{2} t_1^2 + Jt_1 \right\}$$

with the values of A, B, C, D, E, F, G, H, equal to those accepted under 2) above and with $J = GT^{n+2} - FT^{n+1}$.

Second phase.

$$\left\{\begin{array}{l} f_1 = f_0 = a_1 - b_1 v_0 = a_2 - b_2 v_0 \text{ at the transition speed of } v_0. \\ r_1 = r_0 = \varphi\,(v_0) \text{ at the transition speed of } v_0. \\ R_1 = R_0 = \Phi\,(v_0) \text{ at the transition speed of } v_0. \end{array}\right.$$

$$\left\{\begin{array}{l} f_2 = f_B = a_2 - b_2 v_B \text{ at the end of the period (speed } v_B). \\ r_2 = r_B = \varphi\,(v_B) \text{ at the end of the period (speed } v_B). \\ R_2 = R_B = \Phi\,(v_B) \text{ at the end of the period (speed } v_B). \end{array}\right.$$

1) *Calculation by a first approximation of the speed* v_B.

Let v'_B stand for the value of v_B obtained by a first approximation with $\alpha = 0$, $\beta = 0$, $\gamma = 0$.

Under these conditions, if we introduce in equation (4) the following values :

$$\begin{array}{l|l|l} \Theta_1 = t_1 & f_1 = f_0 = a_1 - b_1 v_0 = a_2 - b_2 v_0 & \alpha = 0 \\ \Theta_2 = T & r_1 = r_0 = \varphi(v_0) & \beta = 0 \\ T - \Theta_1 = T - t_1 = t_2 & R_1 = R_0 = \Phi(v_0) & \gamma = 0 \\ T - \Theta_2 = T - T = 0 & & \\ v_1 = v_0 & & \\ v_2 = v'_B & & \end{array}$$

so that :

$$A = -\frac{f_0 Q}{T^n}$$

$$B = 0$$

$$C = 0$$

$$D = f_0 Q + r_0 P' + R_0 \pm iP$$

$$E = \frac{g}{1000\,P\eta}$$

$$F = \frac{BT - A}{n+1} = \frac{f_0 Q}{(n+1)T^n}$$

$$G = 0$$

$$H = 0$$

then equation (4) gives us :

$$\begin{aligned} v'_B &= v_0 - E\{- F t_2^{n+1} + DT - Dt_1\} \\ &= v_0 - E\{- F t_2^{n+1} + D(T - t_1)\} \\ &= v_0 - E\{- F t_2^{n+1} + D t_2\} \\ &= v_0 - E t_2\{- F t_2^{n} + D\} \end{aligned}$$

with values of D, E, and F as taken above.

From these we can deduce the approximate values of $f_2 = f_B$, $r_2 = r_B$ and $R_2 = R_B$, i.e. :

$$\begin{array}{l} f_2 = f_B = a_2 - b_2 v'_B \\ r_2 = r_B = \varphi(v'_B) \\ R_2 = R_B = \Phi(v'_B) \end{array} \qquad \text{whence} \left\{\begin{array}{l} \alpha_2 = \dfrac{f_B - f_0}{t_2} \\ \beta_2 = \dfrac{r_0 - r_B}{t_2} \\ \gamma_2 = \dfrac{R_0 - R_B}{t_2} \end{array}\right.$$

2) *Calculation, by second approximation, of speed* $\dot{v}_B$.

If in the same equation (4) we now insert the values :

$$\begin{array}{l|l|l}
\Theta_1 = t_1 & f_1 = f_0 = a_1 - b_1 v_0 & \alpha = \alpha_2 = \dfrac{f_B - f_0}{t_2} \\
\Theta_2 = T & \qquad = a_2 - b_2 v_0 & \\
T - \Theta_1 = T - t_1 = t_2 & f_2 = f_B = a_2 - b_2 v'_B & \beta = \beta_2 = \dfrac{r_0 - r_B}{t_2} \\
T - \Theta_2 = T - T = 0 & r_1 = r_0 = \varphi(v_0) & \\
v_1 = v_0 & r_2 = r_B = \varphi(v'_B) & \gamma = \gamma_2 = \dfrac{R_0 - R_B}{t_2} \\
v_2 = v_B & R_1 = R_0 = \Phi(v_0) & \\
 & R_2 = R_B = \Phi(v'_B) &
\end{array}$$

so that :

$$A = \frac{\alpha_2 t_1 Q}{T^n} - \frac{f_0 Q}{T^n}$$

$$B = \frac{\alpha_2 Q}{T^n}$$

$$C = \alpha_2 Q - \beta_2 P' - \gamma_2$$

$$D = f_0 Q + r_0 P' + R_0 - \alpha_2 t_1 Q + \beta_2 t_1 P' + \gamma_2 t_1 \pm iP$$

$$E = \frac{g}{1000\ P\eta}$$

$$F = \frac{BT - A}{n + 1} = \frac{\alpha_2 Q}{(n+1)T^{n-1}} - \frac{\alpha_2 t_1 Q}{(n+1)T^n} + \frac{f_0 Q}{(n+1)T^n}$$

$$G = \frac{B}{n+2} = \frac{\alpha_2 Q}{(n+2)T^n}$$

$$H = \frac{C}{2} = \frac{\alpha_2 Q - \beta_2 P' - \gamma_2}{2}$$

we thus have :

$$\begin{aligned}
v_B &= v_0 - E \left\{ G t_2^{n+2} - F t_2^{n+1} + HT^2 - Ht_1^2 + DT - Dt_1 \right\} \\
&= v_0 - E \left\{ G t_2^{n+2} - F t_2^{n+1} + H(T + t_1)(T - t_1) + D(T - t_1) \right\} \\
&= v_0 - E \left\{ G t_2^{n+2} - F t_2^{n+1} + H(T + t_1)t_2 + Dt_2 \right\} \\
&= v_0 - E \left\{ G t_2^{n+2} - F t_2^{n+1} + H(T + T - t_2)t_2 + Dt_2 \right\} \\
&= v_0 - E \left\{ G t_2^{n+2} - F t_2^{n+1} + H(2T - t_2)t_2 + Dt_2 \right\} \\
&= v_0 - E \left\{ G t_2^{n+2} - F t_2^{n+1} + 2HTt_2 - Ht_2^2 + Dt_2 \right\} \\
&= v_0 - E \left\{ G t_2^{n+2} - F t_2^{n+1} - Ht_2^2 + (2HT + D)t_2 \right\} \\
&= v_0 - Et_2 \left\{ G t_2^{n+1} - F t_2^{n} - Ht_2 + 2HT + D \right\}
\end{aligned}$$

with the magnitudes of D, E, F, G, and H, as taken above.

3) *Calculation of the distance* l_2 *travelled in the time* t_2.

If therefore in the equation (5) we now introduce the values $\Theta_1 = t_1$, $\Theta_2 = T$, $T - \Theta_1 = t_2$, $T - \Theta_2 = 0$, $\Theta_2 - \Theta_1 = T - t_1 = t_2$, and $v_1 = v_0$, in the same way as $l = l_2$, the said equation (5) gives us :

$$l_2 = v_0 t_2 - E \left\{ - \frac{G t_2^{n+3}}{n+3} + \frac{F t_2^{n+2}}{n+2} + \frac{H}{3} T^3 - \frac{H}{3} t_1^3 + \frac{D}{2} T^2 - \frac{D}{2} t_1^2 + JT - Jt_1 \right\}$$

with the values of D, E, H, F, and G, as taken for sub. 2) above, and $J = G(T - t_1)^{n+2} - F(T - t_1)^{n+1} - Ht_1^2 - Dt_1$.

In conclusion, the distance travelled by the train in course of the first period, during the given time T will be equal to $L = l_1 + l_2$.

(To be continued.)

[625 .28 (.73)]

An evaluation of railroad motive power.

Four fundamentals of the economic value and utility of the motive-power types. – Characteristics of steam, Diesel-electric, and electric locomotives. — Prospective types,

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PART I.

This discussion of the fundamentals of the selection and use of railway motive power is developed largely from studies conducted and results obtained by the New York Central System, particularly those of the past decade, and our relatively near-term plans and objectives for the future evolution of road-service motive power. We shall dispense with statistical comparisons of detailed operating and cost data from other sources because, unless all essential information is available and included for strictly comparable conditions, the results may be confusing and misleading. Recourse to the use of assumed conditions is of little avail in reaching reliable conclusions.

Fundamentals of the problem.

From present experience, it appears logical that judgment of the possible economic value, utility, and attractiveness of new forms of motive power, either now in use or under development, may best be predicated on four basic considerations.

It should be understood that these four fundamentals are not here set forth in absolute order of importance because, while not inseparable, they are closely related and, to a greater or lesser extent, the influence of one may affect the value of one or more of the others.

Availability and utilization. — Availability is here defined as the percentage of total time that a locomotive is available for service; utilization represents the percentage of total time it is actually in operation and is dependent in large part on the arrangement of schedules. As herein applied, availability means also continuity of operation on the road in terms of reduced delays chargeable to the locomotive and the resulting possibilities of progressively shortening schedules without incurring the high and burdensome expenditures necessary for the improvement of trackage installations, including signals and related facilities, to permit increases in permissible operating speeds. Improved rights-of-way and operating practices are essential, but the final results depend also on the locomotive's ability to support the more intensive utilization demands.

The term availability does not here mean the percentage of time available for a given assigned or selected run which daily may require only a portion of each twenty-four-hour period, with ample time remaining for necessary current maintenance attention. This is accented because not infrequently claims for availability up to 100 per cent are predicated on such operations.

Constantly increasing availability of motive power, with opportunity of correspondingly improved utilization, is an

absolute necessity to keep traffic on the rails in the face of the ever-mounting competition of other forms of transportation, usually subsidized, and to build up such traffic. This trend and need cannot be interrupted and must be faced resolutely and realistically.

Costs of ownership and use. — It is a self-evident fact that, under the incentive system of free enterprise, costs which prevent a fair return on the investment are destructive and self-defeating. Obviously, this fundamental requires careful attention.

Engineering and technological progress alone is not enough. Without sound economic and social enlightenment and a revival of the age-old truth that « something for nothing » cannot be obtained, the potential and productive results of such gains can quickly be nullified and the intended purposes defeated. This is a challenge principally to statesmanship, labor leadership, and business to formulate, adopt, and carry out policies known from the lessons of history to be necessary for the continuation of human progress. If the net earnings of the railroads or other legitimate undertakings are destroyed or unduly restricted, the advantages of engineering and related accomplishments, which otherwise would be made available to all, will be definitely limited or completely eliminated.

Capacity for work. — A motive-power unit must possess a reasonable margin of capacity over that necessary to perform the appointed task if such a unit is to contribute to the betterment of rail transportation. To-day, rapid acceleration from rest or back to permissible running speed following slow-downs is a much-desired characteristic in addition to the established capacity of the machine for straightaway running.

Performance efficiency. — Performance efficiency is an important economic consideration. With other essential requirements fulfilled, there is no question but that a major improvement with respect to motive-power fuel economy would be of far-reaching importance to the railroads and to the country.

Steam locomotive development.

To avoid an undesired digression, in describing briefly the characteristics of the more important kinds of motive power, the questions of improved appurtenances and details of design, materials, and contribution, which contribute to the success of the complete locomotive, have been excluded.

Over the years it has been our unceasing determination and practice to advance the design of the reciprocating steam locomotive, not only to achieve progressively better results therewith, but also to enforce constantly higher standards for new and competing forms of motive power which, in turn, has accelerated the development and improvement of reciprocating steam. As a self-contained power plant, it provides horsepower at the lowest initial cost of any type of locomotive now used or under consideration. Much should be done, however, to increase its usefulness by providing greater freedom from failures and detentions en route and reducing time now used for maintenance, servicing, and inspection. Otherwise, it may not be able to compete on its merits with other forms of motive power and will be faced with a restricted use and market.

With these major objectives constantly in view, it has been our endeavour for succeeding reciprocating steam designs steadily to decrease weight per horsepower developed and to increase the steam-generating plant and drawbarpull capacities and overall thermal efficiencies. The mechanical efficiencies have also been increased through the application of roller bearings and by other

means. Improved distribution of wheel loads and progressively bettered counter-balancing for reduction in the effects on rail and roadbed under high-speed operation also have been sought. Long experience has shown that this practice is of great assistance in the maintenance of rail line and surface for good train-riding qualities, even though the road-bed and track structure used has been second to none.

As an illustration of the fact that with good designing the effects of reciprocating overbalance can be well controlled, it may be said that in 1938 a series of slipping tests were conducted on the New York Central over a short stretch of main-line track with 127-lb. rail section on rock ballast. Test runs were made at train speeds varying from 61 to 82 m.p.h. and with maximum slipping speeds of 123 to 164 m.p.h. while working steam. In the tests at the lower speeds no wheel lifting occurred, but in the final run at a revolving speed of 164 m.p.h. the main drivers only lifted slightly, and later examination disclosed

a number of very slight markings on the rails which, however, were without significance and had no effect on the rails or track structure requiring attention by maintenance forces. No damage to the locomotive occurred in any of these tests.

At the same time, in the advancement of reciprocating steam, tender design and capacity have gone forward apace for the lengthening of runs between fuelings and the taking of other supplies. The use of one-piece cast-steel locomotive frames with integral cylinders, cast-steel trucks — leading, trailing, and tender — and integral cast-steel tender frames, the latter installations being of water-bottom design, for definitely reduced maintenance attention, shopping time, and some weight saving, has been in effect for years on all system steam locomotives built and more recently for the trucks of switch and road Diesels. The underframes and trucks on electric locomotives also have been of this design.

These various features and practices have led to higher availability and serviceability with corresponding increases in miles run per year and mileages between classified repairs.

As a condensed and convenient means of illustrating the evolution of reciprocating steam power on the New York Central System during the past two decades, Table I and Fig. 1 are presented. Table I reveals the progressive increase in capacity and size as necessitated by the demands for higher speeds and heavier trains, and the concurrent reduc-

tion in engine weight per indicated horsepower, which has been over 30 per cent in the period. Fig. 1 shows the coincident growth of drawbar pull and horsepower of over 100 per cent and the ascending speeds at which maximum capacities have been attained.

N. Y. C. Class S-1 4-8-4 type.

The culmination of this work to date is represented by the Niagara 4-8-4 type (*). A basic principle in this development was the incorporation of capacity in excess of that required for current work to be performed in order to obtain the greatest possible continuity of operation, reduced time and expense for maintenance, and possible shortening of schedules. That this principle was correct has already been demonstrated by the performance obtained since the engines were placed in regular service beginning in October, 1945.

Boiler maintenance as compared with other types of high-speed design has been almost negligible up to the present time. The first engine shopped for classified repairs after running approximately 255 000 miles required practically no such work except tube removal.

Mileage between tire turnings has averaged about 190 000 with individual engines running as high as 235 000 miles compared with about 100 000 miles heretofore. This high mileage is attributed to the high factor of adhesion and the consequent absence of slipping, together with the design of spring equalization system which uses coil springs at the connection with the frames, the lower initial resistance in trucks, and the use of lateral-motion devices on front and intermediate driving axles, all of which increase the flexibility of the driving machinery and permit automatic adjustment against variations due to cumulative wear.

While of an entirely new design, the conventional fire-tube boiler has been retained, but elimination of the steam dome permitted increased barrel diameter with corresponding increase in furnace volume and gas areas. Shell courses are made of carbon-silicon steel to save weight. The present working pressure is 275 lbs. per sq. in., but the boiler is designed at a minimum factor

(*) Described on page 480 of the September 22nd, 1945, *Railway Age.*

of safety of 4.5 with 290 lbs. per sq. in. pressure, whereas the minimum factor under Interstate Commerce Commission regulations is 4.00.

The front-end arrangement, similar to that on all modern system power, was proportioned from principles developed in a comprehensive series of stationary boiler tests of a kind first undertaken on Class J-1 locomotives in 1937. With the proper relation of exhaust nozzle and stack and the other essential features and dimensions established, this arrangement results in increasing the flow of the gases with no corresponding increase in the quantity of steam, and effects a reduction in back pressure which is reflected in increased cylinder horsepower and overall thermal efficiency. Another advantage is that no change in stack dimensions or reduction in exhaust-nozzle diameter is necessary for winter operation.

Advanced rod design.

While all power developed is delivered through a single pair of main crank pins, the bending strains on these pins are reduced about 50 per cent as compared with the conventional drive by means of an advance design of roller-bearing rod in which the piston thrust is transmitted in a straight line through the main, intermediate, and rear side rods. With the conventional rod arrangement, it would not have been practicable or prudent to attempt delivering the high horsepower of this locomotive through one pair of main crank pins.

Lightened alloy-steel revolving and reciprocating parts have been used with cross-counterbalance to reduce dynamic augment and resulting stresses in rail and roadbed. The reciprocating parts weigh 1 649 lbs. per side and 22.4 per cent are balanced.

Through the use of a new design four-wheel trailing truck, the ash hopper is enlarged to the extent of providing one cubic foot of volume for each square foot of grate area, which is the largest ratio thus far obtained in system locomotives.

Driving wheels are 79 in. in diameter, but the frame is arranged so that wheels ranging in diameter from 79 in. to 75 in. may be used, depending on future requirements for passenger or freight service. One-piece integral cast-steel frames and cylinders with cast-steel trucks and the integral cast-steel water-bottom tender frame previously developed and used on other modern power were applied to these engines.

An open-type feedwater heater and an extra large superheater are installed, both contributing to the overall steam generating capacity and efficiency.

To save weight, aluminium is used for cabs and running boards. Axles are made of carbon-vanadium steel and crank pins of Timken High-Dynamic steel, while main and side rods are manganese vanadium.

Shields were applied at the smokebox sides to neutralize the vacuum effects ahead of the cab, and, due to the arrangement of deck and seat boxes, the degree of visibility from the cab is superior to that of other modern system locomotives having fully satisfactory conditions, regardless of the large diameter shell courses.

Cast-steel pilots and drop couplers, first applied in 1927 to the J-1 class 4-6-4 type and since continued on all modern passenger power for increased safety of movement, were used also on these Niagara type engines.

For increased availability and continuity of operation, an extra large capacity tender of bed type design, now also installed back of the 50 Class J-3 and 15 Class J-1 Hudson type engines, carrying 46 tons of coal and 18 000 gal. of water, was provided, the water capacity being ample when scooping from

track pans. The running gear forming a part of this tender design originated on the Union Pacific but was introduced for the first time on our system as a part of the S-1 development to permit the use of the 4-8-4 wheel arrangement for the locomotive, necessary for high power capacity without exceeding a total wheel base which would allow convenient turning on the 100-ft. tables still in use at several of the important main-line terminal points. Meanwhile, alternate designs of swivel-truck-equipped tenders, to accomplish the same purpose, are now being developed for possible future use.

A specially important feature of these tenders, worked out and perfected on our system, is the tank venting arrangement which allows scooping water at maximum operating speeds and is so designed as to protect from breakage the windows of trains passing on track pans in the opposite direction and at the same time prevent harmful effects to the track ballast due to overflow.

Capacity tests with 79- and 75-in. driving wheels have been made, from which the horsepower characteristics shown in the tables and charts have been derived. These tests were conducted under regular road conditions of operation and indicate what can be obtained daily in actual service.

One S-1 with poppet valves.

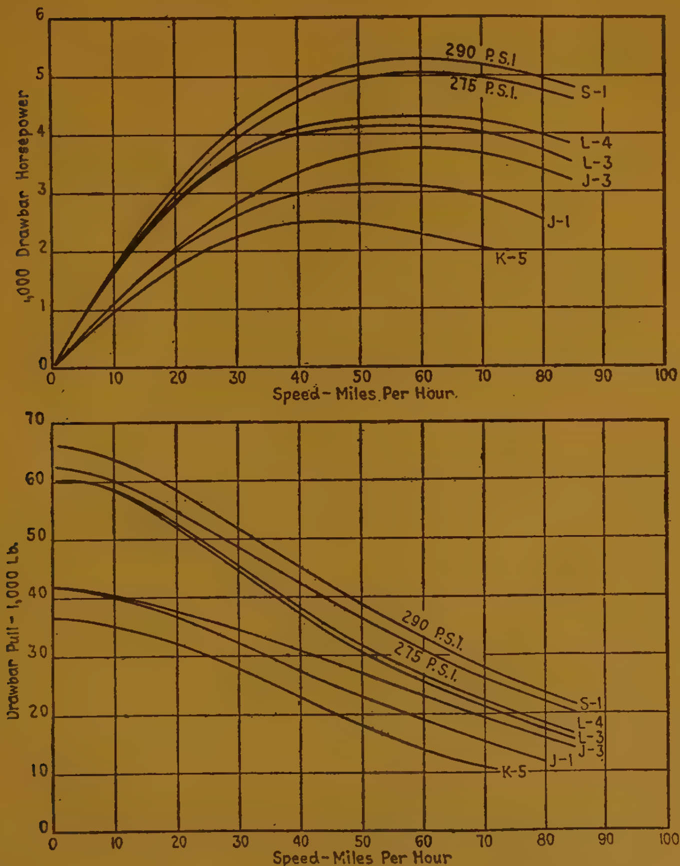
When the New York Central S-1 engines were ordered, it was decided to equip the last one of the lot with the Franklin poppet-valve arrangement having four intake valves 6½ in. in diameter and six exhaust valves 6 in. in diameter per cylinder. With this single exception, the engine is identical with the others, all of which have large-size piston valves actuated by modern valve gear.

Thus, a means for making an exact comparison of capacity, acceleration, and performance for these two kinds of steam distribution was made available, and comprehensive road tests with a dynamometer car are now being conducted with the poppet-valve engines which, with the tests already made of the piston-valve-equipped S-1, will give this exact information. So far as is known, this will be the first such comparison made in the United States on modern high-powered locomotives. When worked up and analyzed later during the present year, the data should provide a basis for definite conclusions.

Other coal-fired locomotives.

In the constant search for better motive power the railroads of the United States have not been content to confine their endeavours and anticipations to the reciprocating steam or the present oil-burning Diesel-electric. The advantages stemming from the continued large-scale use of bituminous coal as a basic motive-power fuel in the United States are numerous and far-reaching, and for the retention and expansion of these benefits several important steps already have been taken and plans for new designs of motive power are under development, such as the Pennsylvania stoker-fired non-condensing steam turbine having mechanical transmission; the three highpowered electric-drive steam-turbine passenger locomotives having a conventional fire-tube stoker-fired boiler now being delivered to the Chesapeake & Ohio, and the gas-turbine-electric using bituminous coal as the basic fuel, two of which are now under development for a group of eastern roads.

Electric locomotives have many advantages. The power being supplied from an outside source provides capacity to the limit of adhesion for short periods, and the characteristics of design permit maximum utilization with more rapid acceleration and better and more reliable overall performance than



Loco. Class	*Dia drivers, in.*	*Working press, lb. per sq. in.*	*Maximum I.HP.*
S-1	79	290	6 900 at 85 m.p.h.
S-1	79	275	6 600 at 85 m.p.h.
L-4	72	250	5 400 at 76 m.p.h.
L-3	80	250	5 200 at 72 m.p.h.
J-3	79	275	4 700 at 77 m.p.h.
J-1	79	225	3 900 at 66 m.p.h.
K-5	79	200	3 000 at 54 m.p.h.

Fig. 1. — Power and traction characteristics of reciprocating steam locomotives.

TABLE I. — **Summary of principal characteri**

Class	K-5	J-1	J-3	L-3
Railroad	N. Y. C.	N. Y. C.	N. Y. C.	N. Y. C.
Type	4-6-2	4-6-4	4-6-4	4-8-2
Service	Pass.	Pass.	Pass.	Frt.
Year first built	1924	1927	1937	1940
Year last built	1926	1931	1938	1942
Cylinders, number, diameter, and stroke, in.	2-25x28	2-25x28	2-22½x29	2-25½x30
Valves :				
Type	Piston	Piston	Piston	Piston
Number and diameter, in.	2-14	2-14	2-14	2-14
Steam pressure, lb, *per* sq. in.	200	225	275	250
Driving wheels, diameter, in.	79	79	79	69
Tractive force, starting, lb.	37 650	42 360	43 440	60 100
Weights in working order, lb.				
On drivers	187 100	190 700	201 500	262 000
Total engine	308 000	358 600	360 000	388 500
Tender (fully loaded)	282 500	305 600	314 300	374 200
Factor of adhesion	4.97	4.50	4.64	4
Grate area, sq ft.	67.8	81.5	82.0	75
Tender capacity :				
Coal, tons	20	28	30	43
Water, gal.	15 000	14 000	14 000	15 500
Maximum cylinder H.P. *and speed* (m.p.h.) *at which attained*	3 000-54	3 900-66	4 700-77	5 200-72
Maximum drawbar H.P. *and speed* (m.p.h.) *at which attained*	2 500-45	3 100-57	3 700-59	4 100-55
Engine weight, lb. :				
Per cylinder H.P.	103	92	77	75
Per drawbar H.P.	123	115	97	95
Piston thrust, max. lb.	98 200	110 400	109 300	127 700
Effective piston thrust load on main crank pin, lb.	98 200	110 400	109 300	127 700
Wheel base, engine and tender, ft.-in.	80-2½	83-7½	83-7½	95-11½

* *Intake* ** *Exhaust.*

iprocating type locomotives.

L-4	S-1	S-1	S-1	S-2
Y. C.	N. Y. C.	N. Y. C.	N. Y. C.	N. Y. C.
-8-2	4-8-4	4-8-4	4-8-4	4-8-4
Frt.	Pass. or frt.	Pass. or frt.	Pass. or frt.	Pass. or frt.
1942	1945	1945	1945	1946
1943	1946	1946	1946	1946
-26x30	2-25½x32	2-25½x32	2-25½x32	2-25½x32
iston	Piston	Piston	Piston	Poppet
2-14	2-14	2-14	2-14	8-6½* 12-6**
250	275	290	275	275
72	79	79	75	79
59 850	61 570	64 930	64 850	61 570
265 800	275 000	275 000	275 000	275 000
397 300	471 000	471 000	471 000	485 000
379 700	420 000	420 000	420 000	406 700
4.44	4.47	4.24	4.24	4.47
75.3	101.0	101.0	101.0	101.0
42	46	46	46	47
15 200	18 000	18 000	18 000	16 000
00-76	6 600-85	6 900-85	6 600-77	Not yet determined
00-60	5 050-63	5 300-62	5 200-61	Not yet determined
74	71	68	71	Not yet determined
93	93	89	91	Not yet determined
132 700	140 000	148 000	140 000	140 000
132 700	70 000	74 000	70 000	70 000
-11½	97-2½	97-2½	97-2½	97-2½

can be obtained with either steam or Diesel-electric. Length and weight per horsepower of the locomotive itself are less than for any other type of modern motive power used and cost approximates that of the modern reciprocating steam for equivalent continuous output rating.

Maintenance outlays are lower and, because less labor is required, future costs should not expand as rapidly as for the other forms of motive power owing to further possible increases in unit labor and material rates and prices. Charges for power should not increase in the future to the same extent as those derived directly from either coal or fuel oil, and rates may become less due to bettered efficiency in generation more than offsetting possible growth in other related expenses.

Because it is a converter of energy and not a prime mover, output is not appreciably affected by mechanical condition nor by manual handling. Running times may more readily be maintained because of the surplus power normally available, which, in turn, provides ability to handle heavier trains and a greater volume of traffic than can be done with locomotives whose outputs are confined to the capacities of their self-contained power plants. Faster acceleration reduces traffic congestion more quickly.

The possibility of the electric power supply causing delay and engine failures on the road is relatively remote, due to the size and capacity of transmission lines and the arrangement of sub-stations and feeders used.

Among the disadvantages of electric motive power are inflexibility and the higher first cost and fixed charges for plant and equipment necessary to generate and deliver this form of energy to the locomotive. With steam or Diesel-electric operation, it is practicable to make use of alternate routes for detouring trains, while the electric is confined to its own tracks, but because of the limited number of occasions that such detouring has been necessary, this is not considered very serious.

Although restrictions in the use of coal-burning steam imposed by municipalities have in some cases resulted in the establishment of electric operation for relatively short distances, which, for the most part, are uneconomical, first cost and resulting fixed charges are the limiting factors for any contemplated electrification of consequence and, unless the density of traffic or other conditions such as line topography are such that an overall return on the investment can be obtained, such operation is not justified.

Diesel electric locomotives.

To obtain a direct comparison of the performance of Diesel vs. the best available steam power in road freight movements, a comprehensive program of test operations was prepared which provided for typical trains and schedules on various divisions and included the handling of both tonnage and fast freight over some of the heaviest grades on the New York Central System.

Train tonnages were predetermined by the use of actual drawbar pull — speed curves for the classes of power involved and the profiles of the territory, and all locomotives were operated at capacity or as closely thereto as was practicable or required under regular conditions of train movements. Equivalent tonnage and number of cars were used with both types, although differing on the several divisions because of varying characteristics with respect to grades, speeds, train loading, and related conditions.

These tests were carried out in the fall of 1944 and in the spring of 1945 under direct supervision of Equipment

Engineering Department representatives, and evaluation of the results revealed, among other basic conclusions, that a three-unit Diesel of 4 500 B.H.P. would handle the scheduled symbol main-line trains over our profile equally as well as the four-unit 5 400 B.H.P. or the latest modern freight steam locomotive of the same power rating as the latter.

In handling main-line passenger trains, the Diesels have shown a high degree of availability and utilization. For the entire year 1946, the average monthly mileage each for the six double units operated was 29 021. As more locomotives are introduced and their use is extended to trains of lesser importance and shorter runs, the availability and the utilization as expressed in miles per month will, of necessity, decrease somewhat, but it is believed that a considerably greater number can be justified, depending on volume of traffic and character of trains operated.

While the Diesels are definitely established in the motive-power field and possess certain important inherent advantages over modern steam, much still remains to be done if they are to continue to meet successfully competition with other forms of motive power.

First cost, weight per horsepower, number of units for a given power output, and overall length must be reduced, and improvements made in power plant and transmission, and long-range repair costs must be kept under control.

Future developments may include mechanical or hydraulic transmission, with a saving in weight and cost, more dependable valve and piston construction, additional fuel and water capacity, and progressive decrease in weight, length, and relative cost per horsepower. As higher-speed engines having overload capacity for short periods are designed and used, with accompanying larger generators and motors for an approach to the performance of the straight electric in the handling of trains, care must be taken that the cost of the higher quality fuel which may be required does not offset the savings realized through weight reduction.

PART II.

As a measure of locomotive potentialities in passenger-train duty, data were assembled for steam, electric, and Diesel-electric operations during 1946 which indicate the comparative degree of availability and utilization for each of these kinds of motive power.

In that year, six 4 000-B.H.P. two-unit Diesels were used regularly on three westbound and three eastbound schedules, one in each direction between Harmon, N.Y., and Chicago, and two between Harmon and Mattoon, Ill., the assigned train-service mileage per day being, respectively, 928 and 1 000. The six locomotives accumulated a total of 2 089 563 miles in 12 months — an average of 29 021 per engine per month, or 954 per day.

Beginning in October of the same year, six of the S-1 class 4-8-4 type steam locomotives, including the one equipped with poppet valves, were assigned to three westbound and three eastbound runs between Harmon and Chicago. Up to the end of November, the accumulated mileage was 314 014, representing an average per engine of 26 168 miles monthly, or 858 per day.

The strike in the bituminous coal mines caused a disruption of this arrangement for 13 days during December, and the engines were assigned to other runs in this period. The total mileage for the three months beginning October 1st was 455 404, or an average of 25 300 miles per month per engine.

Comparative performance.

An analysis of the records of the two Diesels and six Class S-1 for 15 con-

secutive days during October is set forth in Table II, as comparative potential performance on a yearly basis, Harmon to Chicago, under conditions prevailing during favorable weather when the steam locomotives are less susceptible to delays and failures and relative better performance can be expected than for the full 12-month cycle.

The actual average mileage obtained in the entire month of October was 28 954 for the two Diesels and 27 221 for the six steam.

In this operation, both types of power were given the same attention at terminals, but the Diesels were under repair and inspection daily for 21.74 per cent of the time vs. 17.75 per cent for the steam. This accounts for the difference in hours unavailable, Line 6. On the other hand, the time out of service for shopping and periodical required inspections was 3.29 per cent of total hours, Line 1, for Diesels and 7.67 per cent for steam, which produced a lower potential in hours of use for the steam, offset to some extent by the slightly longer steam schedules and resulting in the total hours used, Line 4, and per cent utilization, Line 7, being less for steam than for Diesel. The availability ratio, Line 8, representing the total hours used and hours held waiting, involves the three factors and is almost identical for both classes of power.

For year-'round operation, such performance cannot be expected of the steam power with its inherent disabilities increased during severe winter weather, and the annual mileage anticipated would be less than given in the tabulation. With a conservatively estimated allowance for additional out-of-service time for steam because of increased inherent disabilities during severe winter weather, the following comparative evaluation for potential utilization and availability was revealed from this study :

	Diesel	*Steam*
Annual mileage	324 000	288 000
Average miles per month	27 000	24 000
Utilization, per cent of total annual hours	70.4	63.0
Availability, per cent of total annual hours	74.2	69.0

TABLE II.

Annual potential performance per locomotive — Passenger service.

(Based on 15 consecutive days' service between Harmon, N. Y., and Chicago during October, 1946.)

Line No.		*Diesel*	*Steam*
1	Total hours (365 × 24)	8 760	8 760
2	Hours for shopping and periodic inspections	288	672
3	Assigned hours (1)-(2)	8 472	8 088
4	Hours used	6 292	6 080
5	Hours available, not used	338	573
6	Hours unavailable	1 842	1 435
7	Per cent utilization (4) ÷ (1)	71.8	69.4
8	Per cent availability [(4) + (5)] ÷ (1)	75.7	75.9
9	Mileage operated	329 934	314 694
10	Average miles per month (9) ÷ 12	27 496	26 226
11	Average miles per day (9) ÷ 365	904	862
12	Average m.p.h. (overall schedules)	52.44	51.75

To appraise the possibilities of on-time performance of steam and Diesel-electrics in the handling of passenger trains a study was made of delays en route and at division terminals between Harmon and Buffalo, N.Y., 403 miles, for six S-1 class 4-8-4 type steam and six 4 000-B.H.P. Diesel-electrics in through service on the most important trains during the month of October, 1946.

Steam's handicaps.

Both types of power received the same preferred attention at terminals, and coal of a somewhat higher grade than normal was provided for the steam units. The records of 356 trains, — 179 for the steam and 177 for the Diesels — were examined, with the following average results in minutes :

	Diesel	*Steam*
Gross delay	16.1	21.1
Running time made up .	13.9	17.6
Net time late at final terminal	2.2	3.5
Gross delay chargeable to locomotive . . .	1.3	1.2

Of the total number of trains involved, the number recorded as on time was 134, or 75 per cent, for steam, and 126 or 71 per cent, for Diesels, with average delay chargeable to the locomotive practically negligible for both classes of power. Running time made up nearly compensated for the delays en route and at terminals. The relatively more stable performance of the Diesel was accomplished with the same size trains as for the steam, although rated horsepower was about one-third less, but full capacity of the steam could not be utilized because of train length and other limitations.

Electric locomotives having no self-contained prime mover can be expected to possess inherently greater availability than either steam or Diesel. Experience on electrified portions of the New York Central System and other railroads of the United States confirms this potentiality. Utilization is also higher where the length of runs and the arrangement of train schedules permit. In the short distance of 33 miles between New York and Harmon, with highly congested traffic and many trains arriving at New York during the morning hours, with late afternoon and evening departures, opportunity for intensive use over the 24-hr. cycle is limited and the utilization as a percentage of total time, is low. For the single revenue run in one direction an average of about 1.5 total locomotive-hours is required and an average of about two round trips is made daily, although this is occasionally exceeded.

Steam locomotives labor under several handicaps not applicable to Diesels or electrics, and severe winter weather accentuates these difficulties. Poor coal and low steam pressure, shortage of coal and water, ash-hopper defects and disposal of ashes, heating of friction-type crank pin and other journals, broken machinery parts, failure of piston and valve packing, defective feedwater equipment, and water-scooping apparatus all contribute to failures and delays en route and to late arrivals at terminal points, peculiar to steam only. In some cases these disabilities affect power output. Diesel and electric motive power suffer no diminution of efficiency or capacity under low-temperature conditions.

Out-of-service time.

Considerable progress has been and is currently being made to reduce these difficulties and resulting delays chargeable to modern reciprocating steam. Roller bearings on axles, crank pins and valve gear, the use of alloy steel for machinery parts, refinements in design and insistence on careful workmanship have been of great help, and the provi-

sion of larger ash hoppers has permitted longer runs between servicings. Ash-hopper servicing en route still constitutes a serious bottleneck in the maintenance of fast schedules, particularly during cold weather when it is often necessary to thaw out the hopper in order to discharge the ashes and clinkers. The necessity of refueling at intermediate points and delivering the semi-frozen fuel to the stoker screws, even with the high-capacity steam-operated coal pushers on large-capacity tenders now in use, remains as a serious cause of delay.

Reciprocating steam locomotives also are out of service a greater portion of the total time than the Diesel for shopping and regular terminal attention and maintenance, including governmental inspection requirements. A careful study indicates that over a period of one year, for high-mileage locomotives in Harmon-Chicago service, excluding unforeseen contingencies and operating emergencies, the through-service steam is unavailable a minimum of 28 and the Diesel 12 days. This reduces the potential utilization and mileage of steam, but is inherently characteristic with its high-capacity boiler and relatively complicated reciprocating machinery. Some further improvement can probably be effected in the time required for shopping and for periodic inspection, but it is probable that the Diesel will retain the advantages in this respect.

It is possible that in the future development of reciprocating steam some of the principles and devices proposed for use in the pulverized-coal-burning gas-turbine and steam-turbine motive power may be adapted, particularly with reference to the use of coal fuel and the disposal of ashes and other solid residue of the combustion process.

The Diesel locomotive is subject to certain maintenance difficulties not encountered on steam. The engine itself contains a multiplicity of reciprocating parts in the form of pistons, valves, and related mechanism with which there is a continuing possibility of failure. While not necessarily resulting in locomotive cut-out, sufficient reduction in capacity may take place for delay en route, particularly with the one- or two-unit locomotive. The water and cooling systems

TABLE III.

Annual potential performance per locomotive — Freight service.

(Based on 17 consecutive days' service during October, 1944.)

Line No.		*Diesel*	*Steam*
1	Total hours (365 × 24)	8 760	8 760
2	Hours for shopping and periodic inspections	216	696
3	Assigned hours (1)-(2)	8 544	8 064
4	Hours used	7 219	7 023
5	Hours available, not used	349	252
6	Hours unavailable	976	789
7	Per cent utilization (4) ÷ (1)	82.4	80.2
8	Per cent availability [(4) + (5)] ÷ (1)	86.4	83.0
9	Mileage operated	137 450	118 237
10	Average miles per month (9) ÷ 12	11 454	9 853
11	Average miles per day (9) ÷ 365	377	324

occasionally give trouble and the motors are subject to overheating, but this is relatively rare. Heating of long trains in extremely cold weather has always presented difficulties, and the inadequate equipment heretofore furnished on the Diesels has been a constant source of trouble.

The electric locomotive offers to date the best possibilities for eliminating delays en route due to motive power largely because it does not include a self-contained power plant and related apparatus and, with its greater power input during accelerations, it can readily make up more time following train detentions for various reasons.

Troubles with the electric consist principally of motor flashover, hot motor bearings, and failure of auxiliary equipment. Power supply failure may occur, but experience has shown that little interruption of service is encountered on this account and that this kind of motive power is the best available, or even now contemplated, for minimum delays and maximum utilization. Because it has fewer moving parts, failures on the road are negligible and terminal time for inspection, servicing, and repairs is less than for other forms of power.

Freight service.

The eight Diesel cabs of 1 350 B.H.P. capacity which were acquired in 1944 have been in service continuously since their delivery, most of the time as two-unit 2 700-B.H.P. combinations. In the year 1945 when they were thus used, the total mileage for the four locomotives was 506 608, equal to an average of 10 554 miles each per month.

In July, 1946, two additional cabs were added and since that time the 10 units have been operated as four locomotives, two comprised of three units each and two of two units each. The average miles per locomotive per month in 1946 was about 10 000.

Analysis of the performance of two 5 400-B.H.P. Diesels vs. two modern steam freight locomotives of about the same indicated horsepower capacity, in comparable service, was made for a check period of 17 days in October, 1944. In Table III the results are translated into terms of maximum potential yearly performance.

During the period studied, every effort was made to keep the engines in service and avoid delays at terminals. Favorable weather conditions also contributed to the excellent performance.

Records of actual use over extended periods indicate that for year-'round operation an approximate average monthly mileage of 10 000 for Diesels and 8 500 for steam can reasonably be expected and should be obtained when receiving equivalent attention. With these mileages, the percentages of utilization and availability are evaluated as follows :

	Diesel	*Steam*
Annual mileage . . .	120 000	102 000
Average miles per month	10 000	8 500
Utilization, per cent of total hours	70.1	63.5
Availability, per cent of total hours	73.5	65.7

Proposed new types appraised.

The most attractive present development is the gas-turbine, promising all the advantages of the Diesel, plus the rotating prime mover and low fuel consumption when produced for the successful use of pulverised coal.

For constructive contribution to the possibilities of the pulverized-coal-fired steam-turbine-electric, the high-pressure water-tube boiler offers the advantages of higher steam temperatures and improvement in ash disposal and, if harmful slag deposits can be eliminated there-

from, this kind of locomotive should become a competitor of the Diesel unless total cost is found to be excessive.

The stoker-fired coal-burning steam turbine will also have the advantages of rotatively applied power and the complete absence of cylinders and valves with related reciprocating parts for greater continuity of operation, although, with the conventional boiler and ash pan, it is not expected to equal the Diesel.

Costs of ownership and use.

For analysis, this important fundamental is here divided into its two chief component parts : first costs and resulting annual fixed charges, and operating costs consisting principally of fuel, repairs and crew wages.

First costs. — Prices as of December, 1946, indicate the following approximate relationships among reciprocating steam (taken as 100), the Diesel-electric and the straight electric :

	Complete locomotive	*Per H.P.*
4-8-4 type steam, 6 000 nominal I.HP. (including high-capacity tender) ...	100	100
Diesel electric :		
6 000 B.HP.	214	214
4 000 B.HP.	147	221
Straight electric, 5 000 continuous rated horsepower	114	137

Annual operating costs. — For steam, Diesel, and electric annual operating costs per mile in daily fast, heavy passenger-train movements, through without engine change, Harmon-Chicago, are shown in Table IV, the result of detailed study for comparison of locomotives of equivalent power.

These data apply only to one representative railroad and the class of service indicated. It follows that, of necessity, each railroad should establish its own figures to determine the economic advisability of the different types of power. It is especially noteworthy that an increase of as little as one cent per mile added to 1946 costs for the item of either repairs or fuel amounts to some $ 3 000 per year per engine for the annual mileages given.

For the pulverized or stoker-fired coal-burning steam-turbine-electrics, and the gas-turbine-electric, determination of construction, operating, and overall costs of ownership must await actual experience. These steam-turbine locomotives, because of the absence of all reciprocating machinery, but with the added electrical and coal-handling equipment and the steam-generating plant, probably would approximate, or exceed, reciprocating steam power in maintenance costs, but with some saving in fuel the overall operating cost might be about equal or less. The gas-turbine, if successful, will provide equal power at a large saving in fuel over either steam or Diesel and, having no boiler and little machinery other than for coal processing and handling, maintenance costs should be lowest except for the straight electric.

For both types, the overall economic results will be materially affected by the first cost. If it is found that the first cost of the gas-turbine locomotive may be brought to a figure comparable with the Diesel-electric of equivalent horsepower, it should provide substantial competition for that form of motive power.

Fig. 1 (*) shows the progressive increase in drawbar capacity for reciprocating steam built during the last two decades, from the Class K-5 type, 4-6-2, to the S-1, 4-8-4. These curves, from which the detailed cut-off information has been omitted to simplify the composite chart, were developed during

(*) See part I, page 165.

TABLE IV.

Annual motive-power operating costs.

Line No.		*Steam, S-1, 4-8-4*	*Diesel 4 000 B.HP. (2-unit)*	*Estimated* *Diesel, 6 000 B.HP. (3-unit)*	*Estimated* *Electric, 5 000 continuous HP.*
1	Approximate relative first cost per locomotive (as of December 1946)	100	147	214	114
2	Total annual mileage	288 000	324 000	324 000	324 000
	Cost per Mile				
		$	$	$	$
3	Repairs	.356	.352	.500	.170
4	Fuel	.410	.280	.420	.400 (at .006 per kw. hr.)
5	(3) + (4)	.766	.632	.920	.570
6	Water	.031	.004	.005	.005
7	Lubrication	.011	.030	.045	.011
8	Other supplies	.005	.002	.002	.002
9	Enginehouse expense	.100	.100	.100	.020
10	Crew wages (two men)	.1944	.1979	.2046	.1927
11	Vacation allowance (3 per cent) .	.0058	.0059	.0061	.0058
12	Social Security and unemployment tax (8.75 per cent)	.0175	.0178	.0184	.0174
13	Total per mile (operating)	1.1307	.9896	1.3011	.8239
	Total Cost				
14	Total annual operating cost . . .	325 642	320 630	421 556	226 944
15	Fixed charges (interest, depreciation and insurance)	24 453	38 841	56 640	24 635
16	Total annual cost	350 095	359 471	478 196	291 579
17	Total annual cost per mile . . .	1.22	1.11	1.48	.90*

* Fixed charges for substations and overhead contact system, which represent an important increment of cost for power delivered to the locomotive, are not included in this figure, but, for comparison in the service indicated, would average about 25 cents per mile, making the comparative total $ 1.15. This total, however, makes no provision for the cost of maintenance of substations and overhead contact system, which would amount to a substantial charge against the electric locomotive.

TABLE V.

A Comparison of maximum acceleration characteristics of passenger locomotives (from Fig. 4).

(Level tangent track — Fifteen air-conditioned passenger cars — 1 005 tons.)

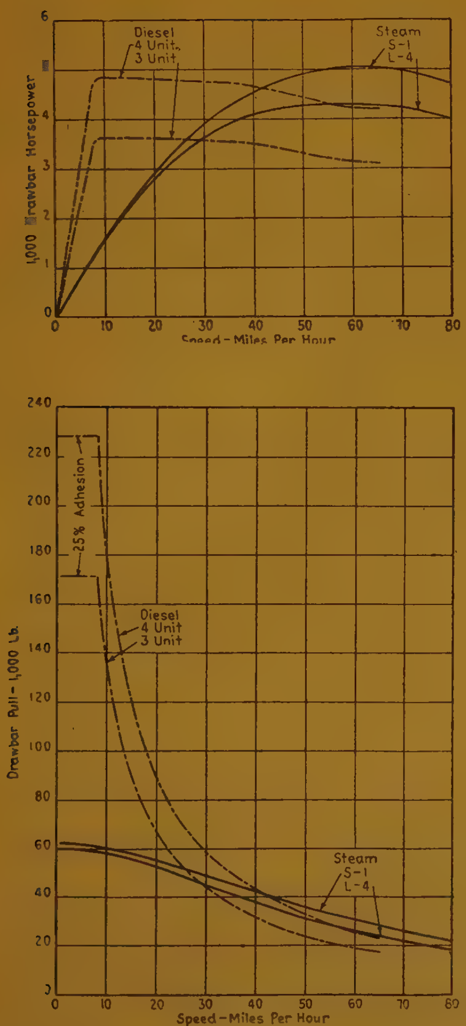
Locomotive type	*Steam*		*Diesel-electric*		*Electric*
Locomotive class	J-3	S-1	2 units	3 units	
Wheel arrangement	4-6-4	4-8-4	2 (0-6-6-0)	3 (0-6-6-0)	4-6 + 6-4
Driving axles, no.	3	4	8	12	6
Horsepower rating	4 700 I.HP	6 600 I.HP.	4 000 B.HP.	6 000 B.HP.	5 000 continuous rated HP.
Working pressure, lb. per sq. in.	275	275	...	...	...
Driving wheel diameter, in.	79	79	...	...	...
Balancing speed, m.p.h.	91	102	84	100.5	105

Acceleration, m.p.h. :	*Min.*	*Miles*	*Min.*	*Miles*	*Min.*	*Miles*	*Min.*	*Miles*	*Min.*	*Miles*
0-35	2.28	0.7	1.58	0.5	1.42	0.5	1.00	0.3	1.08	0.3
0-60	5.00	2.9	3.50	2.1	4.73	3.2	3.06	2.0	2.19	1.2
0-80	9.76	8.6	6.36	5.1	14.17	14.8	6.51	6.1	3.38	2.5
0-100	...	...	16.50	21.3	...	...	26.50	37.8	5.54	5.9
35-60	2.72	2.2	1.92	1.6	3.31	2.7	2.06	1.7	1.11	0.9
35-80	7.49	7.9	4.78	4.6	12.75	14.3	5.51	5.8	2.30	2.2
60-80	4.76	5.7	2.86	3.0	9.44	11.5	3.45	4.1	1.19	1.3

road tests using the dynamometer car under the direct supervision of qualified Equipment Engineering Department personnel, using suitable length trains. A second engine back of the test car was provided for close regulation of trailing-load resistance so as to insure the elimination of practically all acceleration effects during separate runs made to produce the required increments of the complete curve as here illustrated.

With approximately rated boiler pressure and the locomotive as a whole in reasonably good conditions, there is a cut-off for each incidental speed at which maximum power output is made available from a given design, irrespective of whether the capacity of the steam generating plant or the engine cylinders is the limiting factor. Any longer cut-off not only reduces the power output at that speed, but also is wasteful of steam and fuel. At a shorter cut-off, the power output becomes less and it follows that there is a corresponding reduction in steam consumption.

By developing each increment of the pull-speed curve, the succession of cut-off points throughout the range of operating speed to produce maximum power is definitely established and consequently each portion of any such curve can be reproduced at any time in regular train service by using the proper cut-off in relation to speed. This type of operation is utilized for all system reciprocating steam road locomotives, freight and passenger, and is based on the principle of cut-off selection which can be applied directly and conveniently through the guidance of a device known as the Valve Pilot. By means of a duplex gage,



Locomotive.	*Rated H.P.*
Diesel 4 unit	6 000 B.H.P.
Diesel 3 unit	4 500 B.H.P.
Steam S-1	6 600 I.H.P.
Steam L-4	5 400 I.H.P.

Fig. 2. — Comparison of the characteristic curves of steam and Diesel-electric freight locomotives.

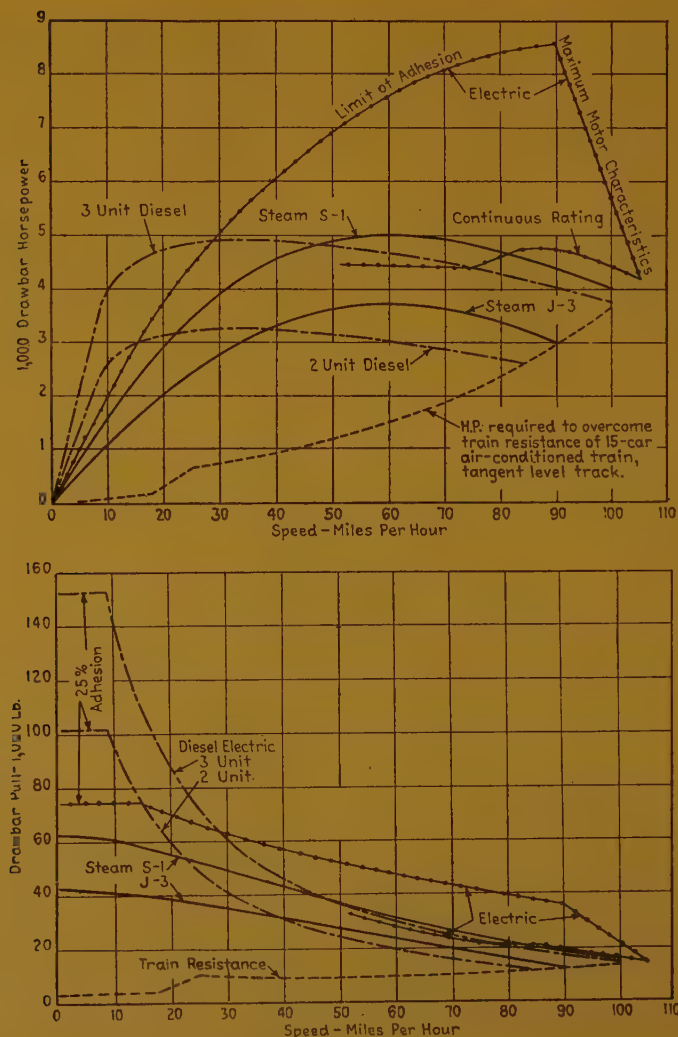
this device indicates the speed and position of the cut-off at all times. Thus, full available capacity may currently be produced while accelerating to running speed and, thereafter, the cut-off may be shortened consistent with load, profile, and speed conditions.

As illustrated in Fig. 1, there has been a gradual increase during the 25-year period from 2 500 drawbar H.P. at a speed of 45 m.p.h. obtained with a K-5 class to 5 050 drawbar H.P. for the S-1, with 275 lb. per sq. in., an increase of 102 per cent. The curve for the S-1 with 290 lb. per sq. in. derived from the test with 275 lb. per sq. in. shows a maximum of 5 300 drawbar H.P., an increase of 113 per cent over the K-5.

Figs. 2 and 3 show, respectively, the power characteristic curves for steam and Diesel-electric freight and for steam, Diesel-electric, and electric passenger power. The relatively constant horsepower of the Diesel engine is shown with the extremely high starting drawbar pull, but rapidly decreasing as the locomotive speed is advanced, until at the range of about 30 to 40 m.p.h. the Diesel and steam of equivalent maximum capacity have approximately the same drawbar pull and horsepower. Thereafter, the steam has the advantage. The electric locomotive, Fig. 3, because power for short-time rating is limited only by the adhesion of the wheels on the rails and the traction-motor characteristics, has higher capacity throughout the entire speed range than reciprocating steam and in excess of the Diesel at speeds over 30 m.p.h.

The electric rated first.

Fig. 3 also includes curves representing the pounds resistance and equivalent horsepower of a 15-car air-conditioned train of 1 005 tons weight on level tangent track, which indicate the balancing speeds for the motive power shown. Fig. 4 illustrates the comparative acceleration characteristics of the three forms



Locomotive.	*Rated H.P.*	*Train resistance.*
Electric	5 000 contin.	15 passenger cars :
		1 005 tons.
Steam S-1	6 600 I.H.P.	85 ft. coupled length each.
Steam J-3	4 700 I.H.P.	Air conditioning and train lighting :
		14 20-kw. generators.
Diesel 3 unit	6 000 B.H.P.	Cut-in speed, 18-25 m.p.h.
Diesel 2 unit	4 000 B.H.P.	Average demand, 60 per cent.

Fig. 3. — Power characteristics of steam, electric, and Diesel-electric passenger locomotives on tangent, level track.

of power, with time plotted against speed and distance, in handling a 15-car passenger train on level tangent track. For convenient reading, Table V is given to show these characteristics up to speeds of 35, 60, 80, and 100 m.p.h. and from 35 and 60 m.p.h. to running speeds of 60 and 80 m.p.h., respectively.

While from the charts the 4 000-B.H.P. Diesel might not be expected to equal the performance of the J-3 steam with its somewhat higher maximum horsepower even though the Diesel accelerates more rapidly up to 60 m.p.h., actually with trains of 15 or 16 modern cars, it has proved more effective in maintaining the fastest schedules between Harmon and Chicago.

Electric power has the advantage over both steam and Diesel in regaining speed from stops or slowdowns because its excess reserve capacity permits accelerating the train at a much faster rate. As a train-handling unit only, the straight electric is the most satisfactory motive power available and, in this respect, it is not expected that new forms herein described, either under development or in use, will exceed it.

Both Diesels and electrics sometimes are run at higher speeds on curves and tangents than reciprocating steam in view of the lower centre of gravity, generally lighter wheel loads, and complete absence of dynamic augment. However, the chief reason for moderate speed restriction of the reciprocating steam is to avoid increased maintenance costs as, if well designed, it is inherently capable of safe and suitable operation at present maximum speeds of other types.

For heavy-grade work, Diesels and electrics have a distinct advantage over reciprocating steam. With the former, the drawbar pull increases rapidly as the speed is reduced. This is true to a lesser degree of the latter. Consequently, slow-speed heavy pulls will not cause stalling when motor capacities are not exceeded, whereas, under similar conditions, the reciprocating engine may slip and stall. The constant torque and the adhesion characteristics of the Diesels and electrics are also of assistance in this respect and, with heavy trains, some double heading or helper service may be eliminated.

Any shortening of passenger-train schedules through increased speeds will require greater power output and higher cost, even though during recent years a definitely downward trend in passenger-train weights has been effected with lightweight streamline equipment of modern design. Regardless of this, more rather than less power is needed, especially for acceleration, if faster schedules are to be provided.

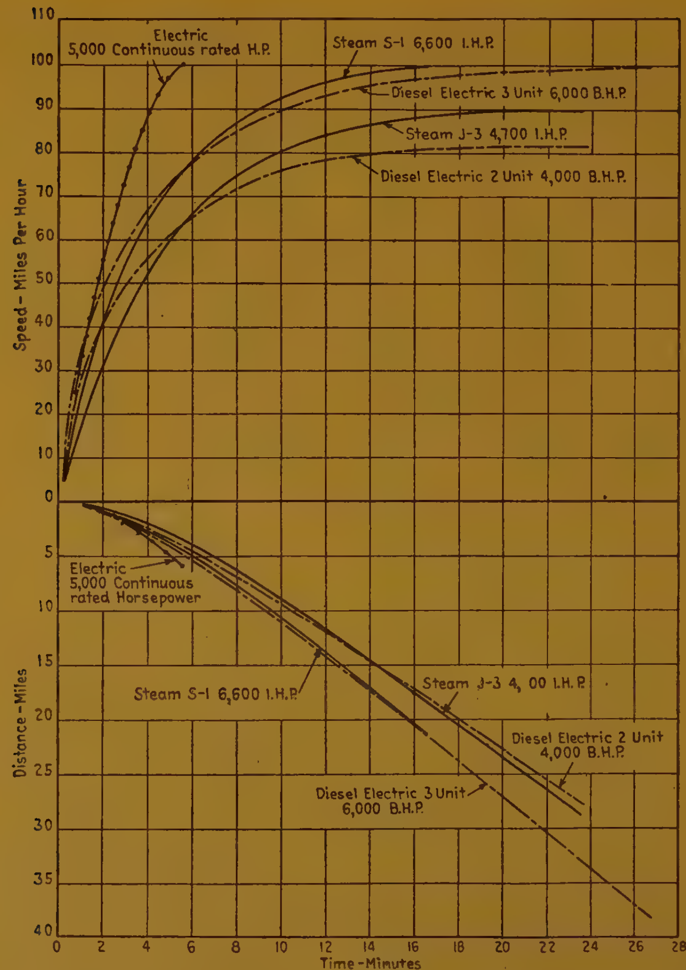
The Diesel's advantages.

The advantages of Diesel-electrics over reciprocating steam may be summed up as follows : (1) little affected by cold weather; (2) lower center of gravity; (3) reduction in track stresses because of lighter wheel loading and absence of dynamic augment, but with this advantage partially offset because of the small-diameter wheels used throughout and the effects of lowered center of gravity; (4) somewhat better riding qualities; (5) less time required for servicing en route; (6) faster acceleration at lower speeds; (7) cleaner operation; (8) higher availability and utilization.

With respect to practically all of the items enumerated, the straight electric is superior to the Diesel.

The Diesel capacity is bound to that of a self-contained power plant, whereas the straight electric is limited by adhesion, and surplus power normally is available at practically negligible additional costs.

The fundamental problem in shortening passenger-train schedules is primarily that of maintaining high average speed, which may most effectively be accomplished through reduction in the



Train resistance.

W. J. Davis, Jr., formulae.

15 cars :

1 005 tons.

85 ft. coupled length each.

Air conditioning and train lighting :

14 20-kw. generators.

Cut-in speed, 18-25 m.p.h.

Average demand, 60 per cent.

Uniform acceleration equation.

$$S = \frac{70}{\text{D.B.P.}} (V_2^2 - V_1^2)$$

$$t = \frac{95.6}{\text{D.B.P.}} (V_2 - V_1)$$

S = Distance, ft.
t = Time, seconds.
D.B.P. = Drawbar pull available for acceleration (Fig. 3), lb. per ton of train.
V_1 = Lower speed, m.p.h.
V_2 = Higher speed, m.p.h.

Fig. 4. — Maximum acceleration characteristic curves of steam, electric, and Diesel-electric on tangent, level track.

number of stops and slowdowns and by reducing the time unnecessarily used with the train at rest. An increase in maximum permissible speed reduces the overall time to some extent, but much more can be gained by raising the overall average speed to the extent found practicable. An analysis of a typical schedule, Harmon-Chicago, showed that the additional time required over the total scheduled running time at permissible speeds amounted to 210 min. — 3 hrs. 30 min. of a total schedule time of 16 hrs. 45 min., or about 21 per cent — divided as follows : decelerating and accelerating to and from slow-downs, 52 min.; decelerating and accelerating to and from stops, 74 min.; timetable allowance for station stops, 62 min.; additional time required for station stops, 22 min.

The number of stops and local speed restrictions and the time thus sacrificed have an important effect on the maintenance of fast competitive schedules and obviously should be kept to the minimum practicable.

Certain of the delays peculiar to steam may be eliminated only by changing to Diesel or electric power, but most of the time lost is independent of the type of motive power as indicated elsewhere in this discussion.

Any reference to the overall thermal efficiency at the drawbar, in the performance efficiency equation usually affords an opportunity for considerable argument. The fact is not questioned that it is highly desirable to improve this characteristic, but it is believed prudent first to review some of the reasons for its relatively low value at the tender drawbar of reciprocating steam.

For reciprocating steam in heavy through service, the necessarily high horsepower requirement is accompanied by elevated combustion rates. This complete power plant, including all auxiliary equipment and its own fuel and water supply, is also handled successively by different engine crews on fast schedules, under widely fluctuating load requirements and, not infrequently, in dense traffic. While the same conditions pertaining to train handling are present also with other forms of motive power, none contains the variables inherent in the production of power from coal-burning reciprocating steam.

The efficiency at the drawbar is affected by the loads on the non-power-producing wheels and those of the tender which, for the latest designs, are the equivalent of two loaded cars having rail weights of 210 000 lb. each.

The retention of simplified design, particularly with respect to cylinders and valve gear, penalizes the thermal efficiency, but repayment is secured and augmented in terms of higher serviceability and reasonable freedom from excessive maintenance troubles and related delays on the road.

Although the overall thermal efficiency of the Diesel locomotive may be four to five times that of reciprocating steam, it should be recognized that without this advantage the cost of Diesel fuel oil would be prohibitive. As a practical fact, the margin in favour of the road Diesel on a fuel basis is of relatively little consequence for equivalent power at current costs for fuel.

Predicated on practice performance of Diesel-electric, straight electric, and reciprocating steam, and thermal analyses of power plants now under development, comparative overall efficiencies may be cited approximately as follows under conditions of full load in train service at a speed of about 65 m.p.h. with fuel of the average quality currently furnished : Diesel-electric, 22 per cent; straight electric, 17 per cent; pulverized-coal-burning gas-turbine-electric (estimated), 16 per cent; pulverized-coal-burning steam-turbine electric (estimated), 10 per cent; modern reciprocating steam, 6 per cent.

TABLE VI.

Relative evaluations of various motive-power types.

Rating	*Availability*	*Overall costs of ownership and use*	*Capacity for work*	*Performance efficiency* — *Overall performance*	*Performance efficiency* — *Thermal efficiency at drawbar*
1st	Straight electric	Diesel-electric Reciprocating steam	Straight electric	Straight electric	Diesel-electric
2nd	Gas-turbine (est.)	Straight electric (see footnote, Table IV)	Gas-turbine (est.)	Gas-turbine (est.)	Straight electric
3rd	Diesel-electric	....	Diesel-electric Steam-turbine (est.) Reciprocating steam	Diesel-electric	Gas-turbine (est.)
4th	Steam-turbine (est.)	....	....	Steam-turbine (est.)	Steam-turbine (est.)
5th	Reciprocating steam	....	....	Reciprocating steam	Reciprocating steam
Unrated	....	Gas-turbine and steam-turbine*	....	....	....

(*) Determination of this value must await actual construction, operation, and maintenance expense.

Summary of evaluations.

The primary mesure of the value of a locomotive is its use. Motive power, when idle because of mechanical defects or other causes, represents a total loss of investment and constant expense and it is continually demonstrated in all service that maximum overall performance efficiency must be sought and secured by the use of units capable of providing consistently high mileage throughout their useful lives.

The relative evaluations given in Table VI are predicated upon ;

1. Locomotives of equivalent power and representing the latest state of the design art.

2. Equivalent through-line passenger schedules and freight operations and efficient use of potential availability.

3. Equivalent maintenance and servicing attention at all times.

4. Presently accumulated knowledge and experience.

5. The exclusion of fixed charges and maintenance expense for motive-power operating, servicing, and repair facilities. Where the use of steam may gradually decrease, some reduction in the facilities required therefore should take place, but as a partial offset to the resulting savings, moderately increasing investment is required in suitable facilities for other forms of motive power as their number becomes greater.

The ratings in Table VI for the gas- and steam-turbine locomotives are speculative and at the present stage are based on design characteristics and possibilities only. For the straight electric, Diesel, and reciprocating steam, the evaluations given are founded on substantial experience and may be considered independently of the others without alterations in the respective sequences of ratings given.

[625 .13 (.44)]

The restoration of the Frejus tunnel,

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(Revue Générale des Chemins de fer, November 1946.)

The Frejus Tunnel (1) connects France with Italy by running under Mount Frejus and is situated on the Paris-Turin line, about equidistant between Chambery and Turin, 100 km. (62 miles). It is the oldest of the great Alpine tunnels. Its construction was begun in 1857 and completed in 1871, when it measured 12 800 m. (13 998 yards) in length, but in May 1881 this figure was increased to 13 668 m. (14 948 yards).

The name of the engineer Savoyard Sommeiller will ever remain associated with this undertaking, where for the first time compressed air boring machines were used.

I. — DETAILS. OF THE TUNNEL. — EARLY DEFECTS. — WAR DAMAGE. — ATTEMPTS AT RAPID RESTORATION.

1. — Nature of the surrounding earth. — Alterations to the original alignment. — Access to the tunnel on the French side. (See Fig. 1.)

From the Italian side the tunnel first traverses some talcose schists, of good formation generally speaking, but liable to split up readily on exposure to the air and fall in slabs of considerable size. It then encounters in turn triassic strata (formed of calcareous earth of compact form, gypsum, dolomite) quartzite and anthraciferous sandstone, again of good condition, then it emerges on the French side in glacial deposits of a somewhat clayish type, containing numerous stones, generally of a quartz character.

As soon as the tunnel was opened, various disturbances of considerable magnitude appeared in the parts of it running through the glacial deposits.

The small Replat tunnel, driven through a seam of this type, suffered so badly in this respect that it had to be abandoned, the line being diverted provisionally to run on an embankment.

Then the Frejus tunnel itself had to suffer, near its mouth, the effects of the unstable nature of the surrounding ground. It was quickly realised that the original alignment could not be maintained and after taking soundings by means of headings and a geological survey conducted by Mr. Lory, Doyen of the Faculty of Science at Grenoble University, a new alignment (AT_3B in Fig. 1) was agreed to in 1879 and brought into use in 1881 in place of the old one (AT_2B).

The new alignment leaves the old 753 m. (829 yards) from the mouth T_3 and entering the mountain practically parallel with the old one on the mountain side, traverses the anthraciferous sandstone and crystalline schists in running under the glacial deposit which caused the trouble in the Replat tunnel. After running some 1 575 m. (1 723

(1) This tunnel apparently is often incorrectly called the Mont Cenis tunnel, by reason of its proximity to the peak of that name traversed by the National Highway No. 6. The tunnel is always so called in Great Britain.

yards) it emerges in a counterfort of deposits (of a length of 160 m. [174 yards]), the surface portions of which were considered by geologists as having arrived at a stable condition. These views have been confirmed in practice, and no important disarrangement has followed, imputable to the action of these deposits since 1881.

In 1939 therefore the Frejus tunnel had three entrances in France, namely :

— the aligning heading (T_1 — Fig. 1) which forms an extension of the main alignment of the tunnel;

— the original mouth of the tunnel, known as the « monumental » mouth (T_2) (1);

— the new mouth (T_3).

In 1943, when Italy ceased to fight on the side of the Germans, the Italian « maquis » blew up the tunnel over a distance of about 1 km. (1 093 yards) on the Italian side, beyond the point where the tunnel proper joins the exploratory heading on the Italian side. The Germans succeeded in repairing the damage in about 2 months.

During the two aerial attacks on Modane (17th September and 11th November 1943) bombs falling in the vicinity of and above the mouth caused the arch to fail, making it impossible to pass through, even on foot.

Finally, in 1944, the Germans, retreating along the Maurienne valley, after having destroyed 31 railway engineering

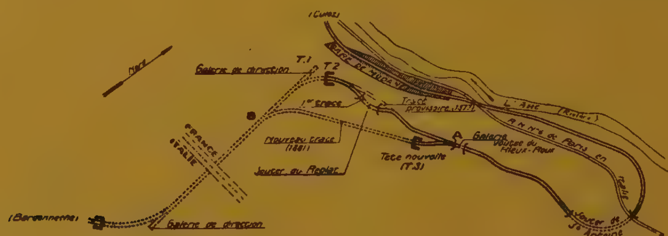

Fig. 1.

Explanation of French terms:

Nord = north. — Galerie de direction = aligning heading. — Nouveau tracé = new alignment. — Soutr. du Replat = Replat tunnel. — Gare de Modane = Modane station. — Tête nouvelle = new tunnel mouth. — Tracé provisoire = provisional alignment. — Soutr. de St Antoine = St. Anthony's tunnel. — Galerie voûtée = covered way. — R. N. No. 6 = National Highway No. 6 Paris to Italy.

2. — The 1939-45 War.

On June 11th 1940, shortly after Italy declared war on France, the French Military Engineers set off the mine at the new tunnel mouth. The damage so produced, which covered some 24 m. (79 ft.) of the side wall of the tunnel on the valley side and a small portion of the arch, was relatively unimportant and traffic through was able to be resumed on August 4th 1940.

works and all the important road structures between St. Pierre d'Albigny and Modane, blew up the new mouth and the entrance to the exploratory heading on the 13th September, the surrounding space remaining heavily loaded with mines and under the fire of the Germans who occupied the crests.

When Italy was liberated, it was learned that the Germans before withdrawing had blown up some 800 to 1 000 m. (875 to 1 093 yards) of tunnel on the Italian side, almost at the same point where the Italian « maquis » had damaged it in 1943.

(1) The original troubles continued to get worse after 1881 but it remained possible to use the tunnel as a footway.

3. — The situation in the early spring of 1945.

All access to the tunnel, whether from the French or Italian side, was effectively rendered impossible, but the damage sustained as far as could be seen from outside was as follows :

On the French side. — At the new mouth the explosion had removed all signs of the actual entrance structure and the retaining walls that went with it. In their place a great clearance was observable among the glacial deposits extending to a height of some 60 m. (197′) above the roadbed of the track and about 100 m. (328′) in length along the centre line of the tunnel (see fig. 2).

It extended over some 30 m. (98′) from the mouth and as far as could be seen the tunnel had been destroyed over a length of at least 40 m. (131′). The « monumental » entrance mouth was intact but 20 m. (66′) further on the arch had been brought down and according to the location of the bomb craters, the damage must have covered some 60 m. (197′).

The surroundings of the exploratory heading led one to believe that the damage inflicted there was of a similar nature.

On the Italian side. — All that was known at the moment was that there was damage of appreciable magnitude at a point about 1 km. (1 093 yards) from the entrance to the tunnel.

Fig. 2. — General view of the fall brought about by the damage done on the 13th September 1944. (On the right the track serving the working site).

4. — Preliminary work. — Attempt to put the tunnel back quickly into working order.

a) *Exploratory heading.*

The first thing to do was to ascertain the exact extent of the damage.

After the mines were cleared away the firm of Borie called in to restore the

tunnel to working order (¹) was given the task of running an exploratory heading almost at right angles to the centre line of the tunnel, breaking into the interior of the latter in rear of the damaged portion. This heading was commenced on the 17th August 1945, and was built to a fairly large cross section (3.15 × 2.50 m. [13′7″ × 8′2⁷/₁₆″]) and thoroughly timbered so as to allow eventually of working from the inside.

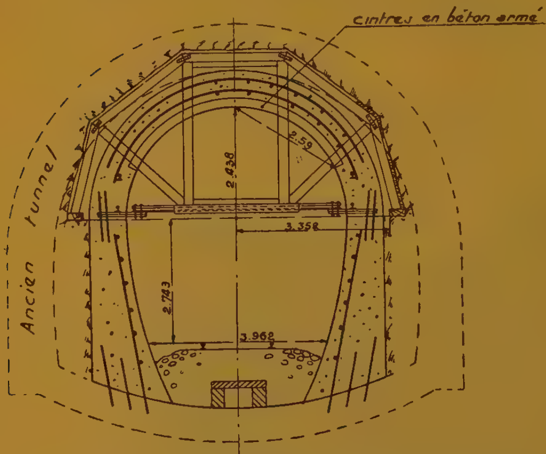

Fig. 3. — Work executed at the « monumental » tunnel mouth. Cross section of the lining proposed to be done by the South Africans.

Explanation of French terms:

Ancien tunnel = old tunnel. — Cintres en béton armé = arch pieces of reinforced concrete.

b) *The help given by the South Africans.*

At the beginning of August 1945, the British General Staff was led to interest itself in the restoration of the tunnel, as it had to make provision for handling between Italy and England a considerable leave traffic, which the blockage of the tunnel compelled it to send through Switzerland.

A South African battalion, composed of engineer troops and miners from the Transvaal, which had repaired numerous tunnels during the war, especially in Italy, was sent to Bardonnechia and Modane, with the object of effecting rapid provisional repairs.

From the French side the proposed way of dealing with the situation was to make use of the old alignment, AT_2B (Fig. 1), after having cleared a way for a track across the collapsed portion behind the « monumental » mouth.

Assisted by a considerable labour force of German prisoners, the South Africans set to work on the debris outside and at the same time, with a view to another attempt being made from inside, commenced to drive a heading at right angles to the centre line of the tunnel, intended to come into it beyond the presumed end of the collapsed portion.

The methods employed were the usual ones : small preliminary top heading, small excavations, then larger ones, formation of arch, running of a layer of concrete over the arch, resumption of work on the abutments and widening out of the side headings (Fig. 3).

In addition to a very highly organised method of working on site, there were two particular processes deserving of mention : the regular use of squared timbering and of arch pieces in reinforced concrete, made in two pieces (Fig. 4), put up joined together and serving as poling boards left in place in the arch.

For reasons which will be explained later, this procedure was abandoned after 10 m. (33′) of arch had been treated with concrete.

On the Italian side, the South Africans at first endeavoured, by making use of the headings used to place mines and not destroyed, and by constructing other exploratory headings, to get round the

(¹) The work was placed in the hands of the two concerns of Borie and Fougerolles, having in mind the important tunnelling operations it was proposed to entrust to the Borie firm. Actually it is this latter firm which has taken on alone the responsibility of equipping and managing the work on site.

part blocked with debris, in order to ascertain the extent of the damage and attack the work in several places.

They brought this preliminary work to a successful conclusion in the middle of September, before on the French side the headings had reached the interior of the tunnel. On September 20th, it proved possible at Bardonnechia to penetrate to the tunnel proper and ascertain the extent of the damage sustained by it.

Fig. 4. — Works executed by the South Africans. Arch pieces of reinforced concrete.

Shortly afterwards, on the Modane side, the exploratory headings driven from the tunnel mouths T_2 and T_3 also penetrated the tunnel and allowed of easier access thereto.

5. — The ascertained extent of the damage.

Italian side. — At a point 745 m. (815 yards) from the entrance to the tunnel, the first fall was found, 100 m. (109 yards) long and estimated to involve 20 000 m^3 (26 159 cubic yards) of debris, with a dome shaped crater 25 m. (82′) high over it and 117 m. (384′) further on a second length of 110 m. (360′) and of approximately 16 000 m^3 (20 927 cubic yards) was found blocked up, also with a crater above 15 m. (49′) deep. These two sections of damaged tunnel were in the talcose schists.

In the rear of the second mass of debris which formed a stopping up of the tunnel, the water which had trickled in had accumulated gradually and made a sheet 3 300 m. (3 609 yards) long and 3 m. ($9'10^{1}/_{8}''$) deep.

French side. — The length of collapsed arch from the new mouth was 50.70 m. (167′). In addition the arch was reduced by some 0.30 m. ($11^{13}/_{16}''$) round the keystone and for the next 20 m. ($65'7^{3}/_{8}''$) sligthly deformed on one haunch.

Four damaged trucks were found amidst the debris and a great many anti-tank mines were scattered in the tunnel which, however, did not appear to have been mined systematically.

At the « monumental » entrance the length of the collapsed portion was 80.30 m. ($263'5^{9}/_{16}''$).

As regards the original exploratory heading the presence of a body of water did not permit of any complete investigation, which in any case would have had no immediate usefulness.

We considered therefore that the time required to restore on the Italian side would be about the same as that needed to put back into a proper state the new mouth on the French side (1) and in these circumstances a preliminary quick solution of the problem lost its attractiveness.

The British General Staff concurred in this view and the works at the « monumental » mouth, which we had undertaken to continue after the South Africans left, were abandoned.

(1) This view proved to be practically correct.

II. — RESTORATION WORK ON THE NEW MOUTH ON THE FRENCH SIDE.

1. — Arrangements adopted.

The arrangements adopted were determined by the nature of the ground which had to be excavated.

The morain deposits at the French entrance to the tunnel had remained stable since 1881; but the damage brought about by the firing of the explosive charges, certainly very considerable, had had a very great effect on them. Inspection of the deposits showed that very probably a portion of the ground alongside the lining of the tunnel had been forced towards the valley and been replaced partially by material coming from the retaining walls and masonry linings, rocky materials and large fragments that had become detached from the deposits [1].

It was necessary to be prepared to go through deposits formed of large pieces, incompletely settled in position, the mass of which would have a strong tendency in parts to drop towards the valley.

It was therefore decided to provide for a heavy lining strengthened again towards the crown (Fig. 5) from 1 to 1.30 m. (3′3³/₈″ to 4′3³/₁₆″) at the keystone, with two stout abutments, 2.40 to 3 m. (6′6¾″ to 9′10¹/₈″) at the roots, strengthened by a crosstie in the form of a 0.70 m. (2′3½″) reversed arch pivot.

The concrete used for the lining was made up as follows : 800 litres (176 gallons) of broken stone, 400 litres (88 gallons) of sand and 400 kgr. (882 lbs.) of artificial cements A. 160-250, save for the abutments where the cement content was 300 kgr. (661 lbs.). The concrete was formed by the vibration process.

The original tunnel mouth was accompanied by a retaining wall of some size, over 18 m. (60′) high. In addition, the slope above the wall had to be packed firm and partially covered with a facing.

It was impossible to contemplate without considerable anxiety rebuilding this retaining wall amidst a mass of unstable deposits; indeed a higher one still would have been necessary by reason of the fall and it would have had to be built in sections. The propping and timbering of those sections, without any serious support from any timbering on the valley side would have been attended with very grave risks, the least of which was the setting up of some movement in the deposits.

It would have been much better, both for the present and the future, to increase the length of the tunnel.

It was decided after a partial clearing of the fallen material in the approach cutting to do this to the extent of 20 m. (63′).

2. — Method used.

The instability of the deposits did not allow, as is often possible when working in new ground, of forming first the lining of the arch and constructing the abutments as a subsidiary process.

It appeared much the more secure way to begin with the abutments, so as to obtain stable foundations for the arch when it came to be built up : this obviously led us to drive a heading for each abutment.

This method had in this particular instance the following advantages :

a) it is convenient in tunnel work to have two levels of headings.

As it is the lowest one which is used the longest, it was the correct course to build the abutments first, since disturbing pressures were feared. The abutment on the mountain side did in fact

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[1] Actually, setting aside a few items of track No. 1 (on the mountain side of the line), no portion of track No. 2 was recovered, either in the tunnel on the valley side nor amongst the ruins of the springing of the arch on that side. The springing itself was completely destroyed and it was necessary to find new foundations at 2 to 3 m. (6½′ to 9′10″) down.

protect the heading on that side; the other allowed of the timbering for the corresponding heading being solidly supported;

b) it was not certain, a priori, that it would prove possible to do all the clearing for the arch and its lining at one operation. The method used would have allowed, should necessity arise, of constructing the span of the arch and part of the springing by starting from headings driven above the abutment headings.

Finally it was settled to proceed on the method illustrated in Fig. 5, the caption of which gives the order of the various operations (clearing and masonry work).

It remained to decide whether the clearing should be effected by working from the outside or the inside.

Without eliminating the possibility of a second attempt from the outside, particularly for the purpose of pushing the work forward, the other method was chosen for the following reasons :

— it is easier, in a heading, to operate on the main body of a fall of material than on its base;

— the clearing of the mines was only effected in August 1945; the delays involved in setting up equipment on site and getting it working did not allow of operations before October. In the neighbourhood in question, at 1 100 m. (3 609 feet) above sea level and with winter temperature of — 15° to — 25° as the normal thing, working outside and the provision of certain supplies (sand and gravel) are practically impossible during a great part of the season. Working

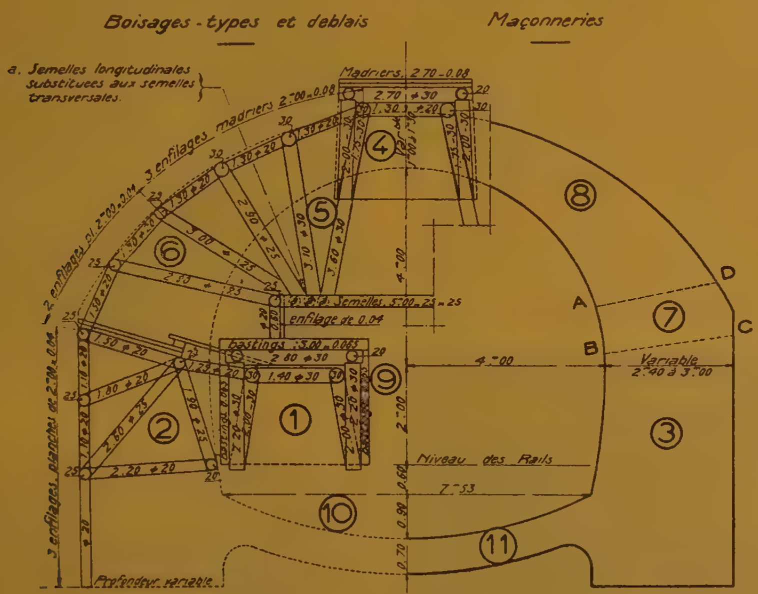


Fig. 5. — Cross section of tunnel.

Stages of the work : 1. Abutment headings. 2. Excavation and trench work for abutments. 3. Concrete for abutments. 4. Top heading. 5. Small excavations in top heading. 6. Large excavations in top heading. 7. Concrete special wall pieces. 8. Concrete arch. 9. Excavating the core. 10. Excavating the invert. 11. Concrete invert.

from the inside, however, would allow of operations proceeding during that period.

3. — Carrying out the work.

As soon as the exploratory heading had been driven on October 4th, the task of taking into the tunnel the gravel, 2 400 tonnes, the sand, 1 700 tonnes, and the cement, 500 tonnes, to be used during the winter, was hurried forward.

a) *Work on the inside* (see Figs. 6 to 12).

After cutting away what could be seen of the damaged wagons and clearing the masonry debris, a screen was erected to hold the foot of the deposits and the lower heading on the valley side was begun.

The clearing of the abutment was effected in alternate sections, starting from the heading, first by making a sinking, then by opening out a trench, in order to make a clearance for the foundations.

The concrete was poured as rapidly as possible and built up as high as could be, so as to cut down the trench and masonry work at the upper level, in con-

Fig. 6. — Looking backwards at the inner works site. Abutment heading valley side. (The movement of the ground towards the valley on the right in the photograph has considerably overturned the timbering.)

Fig. 7. — Inner works. Timbering and masonry for the abutment on the valley side.

Fig. 8. — Inner works. Upper heading. (The setting of the props follows the advancement of the work very closely.)

nection with which serious difficulties were expected.

As soon as the concrete had set sufficiently the timbering of the heading was supported against the abutments to effect a counter-pressure on the deposits.

Fig. 9. — Inner works. Small excavations in the upper heading.

The abutment heading on the mountain side was dealt with somewhat behind that of the other [1], the actual work, however, proceeding in the same manner.

The top heading was begun when the two lower ones had reached a distance of 48 m. (157′5¾″) and 43 m. (141′1″) respectively; this gap in the procedure between the two levels was obviously not compulsory. It would even have been desirable, from some points of view, to reduce it. It was accepted in this instance, owing to lack of skilled operatives to deal with three simultaneous faces, and to enable the difficulties to be allowed for at the upper level to be ascertained more rapidly.

The driving forward of the abutment headings, continued up to a point near where the outside operations were to take place, allowed of working during the bad season in such a way as to accelerate the outside operations in due course.

Shortly after driving the upper heading, the small and large sinkings were undertaken, with 1.50 m. (4′11″) and 2 m. (6′6¾″) between them, and then, with as little gap in the process as the necessities of the situation allowed, the arch raftering was formed and lining with concrete begun.

This was done by serving over the arch rafters in symmetrical fashion. The concreting of the soffit of the arch was preceded by a stage known as the « walling up » which comprised concreting a section of the springing from 1 m. to 1.25 m. (3′3/8″ to 4′1¼″) deep immediately above the abutments (ABCD in Fig. 5). This stage is particularly difficult to effect on account of the position of the concrete at the bottom of the excavated space, amidst the complication of timber props, and also because of the precautions which must be taken to ensure a sound joint between these pieces of wall section and the abutment itself.

Fig. 10. — Inner works, looking backwards. Upper excavation stage. (Right and left clearances; in the distance concrete poured for the last ring and metal arch piece.)

(1) The lack of skilled operatives was the sole reason for doing so.

The concrete on the roof of the arch was put in place by means of 4.50 m. (14′9³/₁₆″) rings corresponding to three spacings of the framings. Keying was avoided by carrying on the concreting without interruption and mixing it by means of the pre-vibrator.

b) *External works.* (Figs. 13 and 14.)

It was possible to carry out only a few preparatory works before winter set

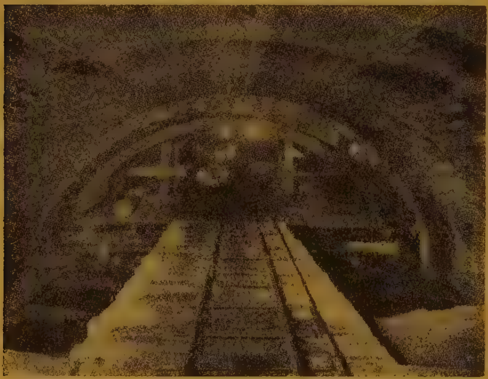


Fig. 11. — Inner works, looking backwards. View of the first rings of the arch. (In the distance the old deformed arch, supported on arch pieces prior to being remade.)

Fig. 12. — Inner works, looking forwards. Site where concrete is being prepared inside the tunnel. (Below the slope giving access to the upper stage is seen the entrance to the abutment heading on the valley side.)

Fig. 13. — Outer works. Entrance to the abutment heading, valley side and to the upper heading.

Fig. 14. — Outer works. Entrance to the three headings and slope giving access to the upper stage.

in; the staff available was concentrated on the work inside the tunnel and on bringing materials together.

By using bulldozers, kindly lent by the South Africans, the roadbed for the tracks was cleared ahead of the fall of deposits.

Using the same machines and handling them with great care, these deposits above the end of the tunnel were evened out. This operation, the result of which is seen in Fig. 14 had as its principal object to reduce the load on the timbering on the mountain side, to increase it on the valley side, and so diminish the slanting thrust exerted against it.

The winter being exceptionally mild, and the difficulty of assembling enough personnel becoming a little less, it became possible on March 25th to begin outside operations.

Having set up a screen to hold the slope, the abutment and arch headings were tackled, the process being the same as that used on the other phase of the work.

4. — Resources used.

The average strength of the staff engaged was 115 workmen, divided among three gangs working 8 hour shifts. In addition some 40 prisoners of war dealt with the handling of materials.

The heavy material comprised :

— two 34 to 45 H.P. compressors for serving the pneumatic hammers and vibrators;

— a concrete mixer of 750 litres (165 gallons) and two of 250 litres (55 gallons) capacity;

— a Montagnet 60 cm. (2′) gauge tractor.

A small workshop was set up for repairing any part of the equipment.

5. — Principal difficulties encountered.

The nature of the ground to be cleared, its want of consistency, and the general trend to move towards the valley, constituted one of the principal sources of difficulty. These determined, as stated above, the choice of the methods to be followed in carrying out operations.

In the first place, it was necessary to have a contractor having at his disposal an excellent staff, able to meet to the greatest possible extent the want of specialised workmen, especially those skilled in timbering.

The timbering for the excavation work was particularly carefully supervised.

Although excellent quality resinous timber, recently felled, was used, and the diameter of the verticals and heads of the stagings and supports was between 0.30 and 0.50 m. ($11^{13}/_{16}''$ and $1'7^{11}/_{16}''$) the timbers were very heavily affected, particularly in the side wall headings.

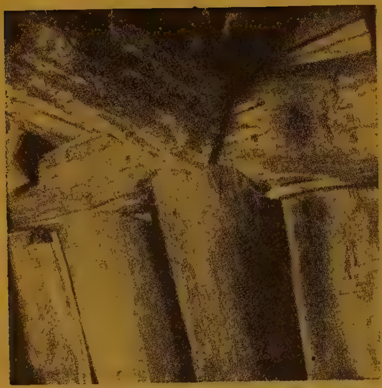
Breakages of the heads and verticals made it necessary to duplicate them in the worst places; Figs. 15 to 22 illustrate some examples of broken or crushed items.

The presence of so much timbering rendered the excavating and masonry work very difficult to carry out.

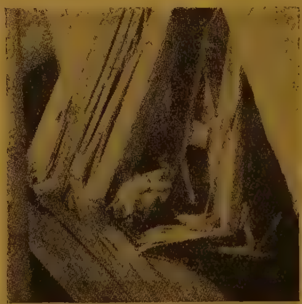
The pressure of the ground caused the headings to settle (the settlement of the top of the heading on the valley side was as much as 0.18 m. [$7^{3}/_{32}''$]) and until the timberings could be supported against the abutments they had a tendency to be forced over, the inclination amounting to 0.17 m. ($6^{11}/_{16}''$) [1]. (This is seen very clearly in Fig. 6).

As soon as the upper heading was able to begin passing material into the abutment headings, the difficulties increased, and led to the modification of the

[1] To ensure the safety of the workmen engaged at the face at the end of the heading in case of a fall occurring behind them, a small heading was driven connecting the two abutment headings, 38 m. (124′8″) from the point where the work was being effected from inside.



Figs. 15 to 22. — Examples of broken or crushed timbering.



sole timbers of this heading and the cutaways at the points of excavation.

Longitudinal sole timbers were substituted for the transverse ones (the setting in place of which was difficult in any case and obliged one to use a mask piece at each staging) so that any accidental load occurring at one point of the top heading would be distributed over several stagings in the lower heading (see Fig. 5).

Finally when the work undertaken from outside was sufficiently advanced, the lower headings were got rid of pro-

Fig. 23. — Abutment heading, mountain side, near completion.

Fig. 24. — Abutment heading, valley side, near completion.

gressively, beginning from inside, by putting stretcher pieces under the centre of the heads of the stagings (Figs. 23 and 24).

The excavating of the upper headings proved difficult, as did that of the abutment headings. However, the building up of the lining followed very soon, within a month at most, after the upper heading was driven.

During this short interval settlements of up to 0.40 m. (1′3¾″) were noted. To meet these the excavating was taken up to 0.70 m. (2′3½″) above the back of the lining, in such a way that, allowing for the thickness of the timber which had of course to be abandoned, the minimum thickness laid down for the lining could be obtained without having to make a « lift » which would have been extremely costly and awkward to do.

The effect on the timbering became extremely noticeable at the break up of the frost and in the zone 25 m. (82′) long situated at some 20 m. (65′7³/₈″) from the point where work was commenced inside.

The amount of timber used gives an idea of the loads and pressures; it amounted to an average of 25 m³ (882 cubic feet), stopping off included, per lineal metre of tunnel and about 0.3 m³ (10.59 cubic feet) of excavation (1).

This represents a particularly high proportion, never before reached in the many tunnels constructed by the P.L.M. Company since the commencement of the century, some of which were certainly difficult to drive.

It may be noted as a matter of information, that for the restoration of the tunnel towards the mouth on the Italian side, not more than 5 m³ (176 cubic feet) of timber needed to be used per metre of tunnel.

Nothing particular needs to be said about the concreting work, except to remark on the difficulty arising from the presence of so much timbering.

We may however point out that the most difficult stage of all, the making of

(1) The so-called « normal » arrangements of timbering are shown in Fig. 5. Their cost was included in the price charged for excavating. Additional timbering was paid for to the contractor at prices fixed by a schedule.

the special wall pieces at the base of the crown of the arch, could have been simplified and even got rid of in another case of the kind by raising from 0.80 m. to 1 m. (2′7½″ to 3′3³/₈″) the level of the abutment headings. The trench work for the foundation of the abutments would have had to be correspondingly deepened, but experience has shown that such an operation presents no special difficulties.

It is intended to take advantage of this in the work of restoring the tunnels on the Nice to Coni line.

In normal times, a great part of the labour available in the Maurienne district comes from Italy. This was lacking however by reason of administrative and political difficulties which put obstacles in the way of obtaining any number of the specialised workers available on the other side of the Alps. The contractor had considerable difficulty in this deserted district in getting together, keeping together, and training the necessary skilled personnel.

Finally the presence of unexploded mines and bombs among the fallen deposits could not be ignored, and the workers had to be on their guard at every instant.

In fact, a pile of shells set ready for firing were found among the earth which caused a certain delay in carrying out the work.

6. — Cost of the work.

The cost amounted to some 45 millions, or about 650 000 francs per lineal metre of tunnel. This figure may be compared with that applying to the Mourepiane tunnel, on the connecting line joining the new dock basins of the port of Marseilles to the Estaque to Joliette line; this tunnel, now under construction, driven through relatively easy ground, is costing 105 000 francs per lineal metre at ordinary labour rates and comparable prices for material.

III. — REMARKS ON THE TIME TAKEN TO CARRY OUT THE WORK.

It took about 10 months to restore a length of tunnel of some 70 m. (229′8″). This delay was attributable to having to make two attempts. It could have been reduced to from 3 to 4 months if the season of the year had allowed of making two starts on the work simultaneously and if enough men to do the timbering had been available.

Time could also have been gained by doing the excavation of the cutting at the tunnel mouth by a bulldozer or an excavator so as to reduce the amount by which the tunnel had to be lengthened and by excavating and timbering in the tunnel less cautiously than was in fact the case.

No doubt one would have been willing to risk doing so had military necessity called for it, but as this was not the case it was held that the safety of the workers ought to be the first consideration.

It was also thought that a fall in a heading, above all in the upper level, would have led to a delay and an expense out of all proportion with the small saving in time that would have been realised.

The elder of the authors of the present article has been engaged on a sufficient number of civil engineering works — especially the building of tunnels — to know that in difficult cases the surest guarantee of success is to avail oneself of the services of a first class contracting firm, having at its disposal a thoroughly skilled staff of workers and foremen.

We are pleased to be able to say that this was the case at the Frejus tunnel works, and the experience of the General Manager of the undertaking, Mr. BORIE and that of his representative Mr. GERVAIS and his works supervisors proved extremely valuable in the very particular circumstances obtaining there.

[385 .113 (.492)]

Report on the year 1946 of the Netherlands Railways Cy.

The Netherlands Railways have just published the report on the year 1946 which was presented to the General Meeting of shareholders on the 19th July 1947. This report is interesting from more than one aspect, so that we think our readers should be informed of the most salient features.

The operating resulted in a net profit of nearly 84 millions. This result, which shows that this railway is in a privileged position compared with others, is due to the increase in the traffic. It is however not exceptional for the Netherlands Railways to show a profit in their balance sheet. Since 1926 there have only been two years, 1936 and 1945, when there was a deficit. Leaving aside the war years, the operating coefficient of 68.8 is the most favourable since 1926.

On the profit and loss account, after deducting 7 millions of interest, setting aside 69 millions for the sinking fund and contributing 7.5 millions to the State exchequer, there is a final profit of 400 000 florins. This represents 4 % on the capital stock held by the State.

The report stresses the fact that the results briefly analysed are a complete answer to the statements often made in certain sections of the press to the effect that the Netherlands Railways are struggling with a large financial deficit which the State has to make good by means of grants from or taxes on road transport.

Nor can it be argued from the fact that profits are distributed to lower the rates. It suffices to say that if the profits represent 4 % of the capital stock, the proportion falls to ½ % when compared with the net operating receipts, and to 2/1000ths if the passenger traffic receipts are used as the divisor. The effects of this factor on the rates is therefore extremely small.

The passenger traffic was particularly heavy during 1946. It is estimated that the number of passengers, which reached 174 millions, was 2.15 times that of 1938. As the difficulties experienced at the beginning of the year decreased, the traffic continued to grow steadily. Certain bus services which were set up as a temporary relief could soon be dispensed with. Altogether the passenger trains covered 20 million kilometres (12.4 million miles), the goods trains more than 9 millions (5.6 million miles), i.e. a sum total of nearly 30 million train-kilometres (18 million train-miles). Out of the total receipts of 238 millions, the passenger services accounted for 198 millions and the goods services for 40 millions. Military transport contributed a further 22 millions to the receipts.

Amongst the chief obstacles in the way of normal operating, mention must be made of the important engineering works. Of the 11 important bridges still unusable on the 31st December 1945, 7 were either provisionally repaired during 1946 or replaced by a temporary bridge. Nearly all the small bridges were restored either provisionally or definitively. At the same time the signal boxes were repaired (98 in all), the block system (1 524 km. [947 miles] in service out of 1 966 [1 214 miles] in September 1944), the automatic signalling equipment of unprotected level crossings (22 restored out of a total of 42) and the telegraph lines.

A very serious source of trouble was the shortage of rolling stock, both loco-

motives and waggons and coaches. Compared with the situation on the 17th September 1944, the number of coaches had been reduced from 1 572 in service to 554 at the end of 1946; the number of waggons was halved. Locomotives were not so badly hit. Coaches were repaired by making use of those beyond repair for spares. The lack of machine tools greatly hindered the activity of the repair shops. Nevertheless production reached 72 % of that of 1938 and 1939 at Amersfoort, and 86 % at Blerick. Repairs were also carried out by industrial firms. The railway purchased English locomotives lent to the Allies in 1945 and also put into service 44 Swedish and 27 Swiss locomotives, in addition to 7 diesel-electric shunting engines and two others for goods trains.

As far as electric traction is concerned, by the 31st December 1946, 286 km. (178 miles) of line had been restored to traffic including the Amsterdam-Utrecht section equipped in 1946, compared with 566 km. (352 miles) in operation on the 17th September 1944.

At the end of 1946, the system included 3 542 km. (2 200 miles) of lines, 2 235 (1 388 miles) of which were single track and 1 307 (812 miles) double track. The total staff amounted to 39 976 employees (i.e. 11.2 per km.). The staff expenditure represents 70.9 % of the operating costs. In 1945 this ratio was higher : 85.4 %.

At the end of 1946, the period of provisional restoration was more or less at an end, to be succeeded by definitive reconstruction. Such work is however seriously impeded by the lack of materials and capital. The Management hopes with the assistance of the Government to be able to actively pursue the carrying out of many tasks of reconstruction in order to maintain a reasonable transport system in the country.

The general impression given by an examination of the year 1946 is encouraging. The financial situation is healthy and thanks to the foresight of the management, the future can be faced with confidence.

E. M.

NEW BOOKS AND PUBLICATIONS.

[385 .09 (.66) & 623 (.66)]

TURNER (Major C. R.) E. D., A. M. Inst. Mech. E., General Manager and Harbour Authority. — **Gold Coast Railway and Takoradi Harbour, 1939-1945. War Activities.** — A brochure (4¾" × 7") of 16 pages and numerous tables. — 1946, Accra, Government Printing Department. (Price : One Shilling.)

The railways and harbours of countries fighting against the Axis had to deal with an exceptional volume of traffic. Often, they had to deal not only with a sudden and considerable traffic but also with a large amount of unusual transport. These additional burdens were increased by work in connection with installations and manufactures largely outside the normal activities of a railway. It is therefore not astonishing that such transport undertakings as the Gold Coast Railway and Takoradi Harbour had much to cope with in view of the extent and diffusion of the theatre of war and also on account of the relative proximity of North Africa and the Middle East. This little pamphlet is however extremely revealing. It gives us in a few pages many details of the unimagined difficulties which had to be solved with fewer facilities available than in times of peace.

Amongst the circumstances which aggravated the burden of the railway, was first and foremost the lack of petrol. Nearly all the road traffic had to go by rail. There were in particular large scale troop movements, especially when leaving to East Africa for the Abyssinian campaign and later on to the Far East.

To support the North Africa campaign the Royal Air Force established a base at Takoradi. This involved the unloading of hundreds of packing cases containing aircraft and all their associated equipment : hangars, spare parts, provisions, munitions, etc.

The working of the bauxite mines, in addition to involving a large amount of traffic, necessitated the construction of a new line 74 km. (46 miles) long. The extensions to the services involved thereby as well as the running of night services led to serious labour problems. Intensive training was necessary. There were also gaps in the managerial staff owing to men being called up.

Nearly all the installations had to be extended, not only on the railway, but also at the harbour as well as its equipment.

The rolling stock, both locomotives and vehicles, had very intensive use, as it was not until much later, in 1944, that new stock could be bought.

The above are brief indications, of a summary nature, gleaned in the course of reading this very interesting little work. They will suffice to show the amount of courage and ingenuity displayed by the staff of these two transport undertakings in the common cause.

The author has appended some statistical data relating to the working years from 1939-40 to 1944-45, which makes it possible to appreciate the facts given in the text.

E. M.

[656. (.469)]

LAPA (J. F.), Doctor of Economic and Financial Science. Head of the Commercial Department of the Portuguese Railways. **Transportes terrestres. Concorrência e coordenaçâo.** (Overland Transport. Competition and co-ordination). One volume ($6\frac{1}{4}$" × 9") of 336 pages. — 1946, Lisbon, Grafica Santelmo, Rua de S. Bernardo, 84.

The author after having examined the universality of the problem of competition between the different methods of transport and the specific characteristics of competition between methods of transport in Portugal, considers, on the basis of the official statistics, the pernicious effects of such competition and throws into relief the extent, acuteness and topicality of the problem. These characteristics enable him to deduce the imperative need for co-ordination as the only solution able to prevent the harmful effects of interference with the functions of each of these methods of transport.

He goes into the matter minutely and compares the legislation governing the commercial operation of the railway and road transport, the system of taxation, the plan of the railway system and the plan of the road system, suggesting in conclusion the drawing up of a « plan for overland transport ». In the chapter devoted to a detailed comparison of the railway rates and prices charged by road hauliers, the author defends the thesis that the former, owing to their eclectic basis, are truly instrumental in the economic and social policy, whereas the latter, as now operated, have no interest in the national welfare, and he explains the consequences of this difference.

Finally, basing his arguments on a very extensive bibliography, the author defines a method of co-ordination under the fourfold aspect of the legislation governing the commercial operation, taxes, a plan for overland transport, and finally the rates, defending contrary to the general tendency the extension to the road transport industry of the legal and economic principles on which railway transport is based, as the sole way to satisfy the real economic principles. In order to make certain of the success of such a system the author recognises the necessity of getting rid of the present haphazard system of hauliers and suggests to his readers two solutions, one being the integration of the transport industry and the other the concentration of road transport, explaining the reasons why he prefers the latter solution.

Finally the author proceeds to examine critically the proposed law on co-ordination presented by the Portuguese Government to the National Assembly in 1945 and the resulting law.